DEVELOPMENT OF A CONTINUOUSLY OPERATED SOLAR POWERED REFRIGERATION SYSTEM

BY

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A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

JUNE 2011

In the name of Allah, the Most Gracious and the Most Merciful

KING FAHD UNIVERSITY OF PETROLEUM AND MINERALS

DHAHRAN 31261, SAUDI ARABIA

DEANSHIP OF GRADUATE STUDIES

This thesis, written by MUHAMMAD UMAR SIDDIQUI under the direction of his thesis advisor and approved by his thesis committee, has been presented to and accepted by the Dean of Graduate Studies, in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in MECHANICAL ENGINEERING.

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Dedicated

to

My Beloved Parents

ACKNOWLEDGMENTS

All praise and thanks are due to Almighty Allah, Most Gracious and Most Merciful, for his immense beneficence and blessings. He bestowed upon me health, knowledge and patience to complete this work. May peace and blessings be upon prophet Muhammad (PBUH), his family and his companions.

Thereafter, acknowledgement is due to KFUPM for the support extended towards my research through its remarkable facilities and for granting me the opportunity to pursue graduate studies.

I acknowledge, with deep gratitude and appreciation, the inspiration, encouragement, valuable time and continuous guidance given to me by my thesis advisor, Dr. Syed ◌A. M. Said. I am highly grateful to my thesis co-advisor Dr. Maged A.I. El-Shaarawi for his valuable guidance, suggestions and motivation. His strong support and co-operation towards this research was a boon to me. I am also grateful to my Committee member, Dr. Habib Abulhamayel for his constructive guidance and support.

My heartfelt thanks are due to my parents for their prayers, guidance, and moral support throughout my academic life. My parents' advice, to strive for excellence has made all this work possible.

Special thanks are due to my senior colleagues at the University for their Help and prayers.

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THESIS ABSTRACT (ENGLISH)

NAME: **MUHAMMAD UMAR SIDDIQUI**

TITLE: **DEVELOPMENT OF A CONTINUOUSLY OPERATING SOLAR POWERED REFRIGERATION SYSTEM**

MAJOR FIELD: **MECHANICAL ENGINEERING**

DATE OF DEGREE: **RAJAB 1432 (H) (JUNE 2011 G)**

The air-conditioning and refrigeration systems in the KSA consume more than 60 % of the electrical energy consumption of the building sector. Most of the used systems are of the vapor-compression type. Using solar energy to power such systems will save a large amount of electrical energy that can be utilized by the production sectors such as the industry. Hence, the focus of the proposed thesis topic is to develop a continuously operating fully solar- powered absorption refrigeration system that can provide a 24-hour service of refrigeration and/or air-conditioning. The development will include an in-depth review of the design and operation of the conventional and solar-assisted absorption refrigeration systems, analysis of refrigerant-absorbent solutions for solar-powered systems, coming-up with new alternative designs, detailed thermodynamic analysis of some of the new alternative designs, selection of the most suitable alternative design and finally preparation of the working drawings for a model of the selected design.

MASTER OF SCIENCE DEGREE

KING FAHD UNIVERSITY OF PETROLEUM AND MINERALS

Dhahran, Saudi Arabia

THESIS ABSTRACT (ARABIC)

ملخص الرسالة

الاسم: محمد عمر صديقي

عنوان الرسالة: تطوير نظام تبريد يعمل باستمرار عن طريق امتصاص الطاقه الشمسيه.

التخصص: الهندسة الميكانيكية

تأريخ التخرج: رجب 1432 هـ - (حزيران 2011 م)

أنظمة تكييف الهواء والتبريد في المملكة تستهلك أكثر من 60 ٪ من استهلاك الطاقة الكهربائية في قطاع البناء. معظم الأنظمة المستخدمة هي من نوع ضغط البخار. يؤدي استخدام الطاقة الشمسية لنظم الطاقة مثل حفظ كمية كبيرة من الطاقة الكهربائية التي يمكن استخدامها من قبل القطاعات الإنتاجية مثل الصناعة. وبالتالي ، فإن التركيز على موضوع الأطروحة المقترحة لتطوير نظام تشغيل تبريدي يعمل بامتصاص الطاقه الشمسيه التي يمكن أن توفر نظام خدمة 24 ساعة للتبريد و / أو تكييف الهواء. كما سيتضمن المشروع إجراء استعراض متعمق لتصميم وتشغيل أنظمة التبريد التقليدية و انظمه امتصاص الطاقة الشمسية ، تحليل محاليل التبريد للأنظمة التي تعمل بامتصاص الطاقة الشمسية، بالتالي تصاميم جديدة بديلة ستدرس، والتحليل الحراري لبعض هذه التصاميم البديلة الجديدة، ومن ثم اختيار التصميم البديل الأنسب، وأخيرا إعداد الرسومات التنفيذية للنموذج للتصميم المختار.

شهادة ماجستير علوم

جامعة الملك فهد للبترول والمعادن

الظهران ، المملكة العربية السعودية

CHAPTER 1

INTRODUCTION

The excessive demand for air conditioning in the Kingdom of Saudi Arabia is a direct result of the extreme temperatures during summer, when the ambient temperature frequently reaches 46 deg C at night [1]. Thus it is imperative to use refrigeration and air-conditioning in all fields of life. The application of refrigeration and air-conditioning is mostly in the building sector i.e. commercial, industrial and residential buildings. Presently almost all the cooling produced in the Kingdom is by means of vapor compression systems. The compressor of these vapor compression systems are directly run by the electrical energy that is generated by burning fossil fuel. This electrical energy is high-grade energy and it can be effectively utilized in high value applications such as industrial purposes. About 65% of the electric energy generated in Saudi Arabia is used for operating buildings, which is consumed by air conditioning [2]. Therefore, it creates the need for the development of a refrigeration system that may run on an alternative source of energy so that electrical energy from the fossil fuels may be employed in the production sector rather than being utilized in the consumption sector such as comfort conditioning. Such systems will significantly share the load of electrical energy generated by burning fossil fuel, thus helping in the reduction of carbon emission, hence reducing environmental pollution and global warming effects.

Out of the various renewable sources of energy, solar energy proves to be the best candidate for Refrigeration and air conditioning because of the coincidence of the maximum cooling load with the period of greatest solar radiation input. Solar energy can be used to power a refrigeration system in two ways. First way is that, solar energy can be converted into electricity using Photo-Voltaic Cells and used to operate a conventional vapor compression refrigeration system. Second way is that, solar energy can be used to heat the working fluid in the generator of a vapor sorption (absorption or adsorption) refrigeration system. Kim and Ferreira [3] made a comparison between solar electric and solar thermal refrigeration systems both from the point of view of energy efficiency and economic feasibility. Solar electric refrigeration systems using Photovoltaic appear to be more expensive than solar thermal systems. Enibe [4] determined that the major problem with PV-powered refrigeration is the system cost. It was found that although solar cell efficiencies have raised from about 7 to 13% [5], and the unit costs have fallen [6], the overall system cost are still above the costs of ordinary vapor-compression refrigerators. Hence, solar absorption refrigeration systems can play an important role in the development of small communities especially those living far from the electrical network.

1.1 Reason for focusing on Vapor Absorption Systems

Most of industrial process uses a lot of thermal energy by burning fossil fuel to produce steam or heat for the purpose. After the processes, heat is rejected to the surrounding as waste. This waste heat can be converted to useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle. Similarly,

solar energy can also be used for this purpose. Electricity purchased from utility companies for conventional vapor compression refrigerators can be reduced. The use of heat operated refrigeration systems help reduce problems related to global environmental, such as the so called greenhouse effect from CO, emission from the combustion of fossil fuels in utility power plants.

1.2 Aqua-Ammonia Vapor Absorption Refrigeration System

A typical aqua-ammonia vapor absorption system can be subdivided into basic components and performance enhancers. Basic components are those components, which are necessary for the operation of the refrigeration system i.e. without it the refrigeration system will not function at all. However, the performance enhancers are those components, which are not essential for the operation of the refrigeration system, although their presence may substantially improve the performance of the system. Generator, Absorber, Condenser and Evaporator are the basic components however, Dephlegmator, Rectifier, Vapor Liquid Heat Exchanger (VLHE) and Liquid-Liquid Heat Exchanger (LLHE) are the performance enhancers of the typical aqua-ammonia vapor. In this thesis, it is required to utilize solar energy to energize the generator of the vapor absorption system, hence obtaining a continuous cooling.

CHAPTER 2

LITERATURE REVIEW AND OBJECTIVES

2.1 Literature Review

Several researches in this field have been done and a lot more is still undergoing. For instance, Wang et al [7] discussed the various typical systems with small scale for potential residential applications in which the working principals, system suitability for solar cooling, performance, maintenance and economic viability have been reported. Chidambaram et al [8] reviewed research articles in the field of solar cooling techniques, solar collectors, storage methods and their integration, along with performance improvement studies reported using thermal stratification and cascaded thermal storage systems. Cerezo et al [9] used ammonia water as the working fluid to look at how absorption takes place in a plate heat exchanger operating under typical conditions of absorption chillers, driven by low temperature heat sources. It was determined that the increase in pressure, solution and cooling flow rates positively affect the absorber performance, on the other hand an increase in the concentration, cooling, and solution temperature negatively affects the absorber performance. A. Sathyabhama and T.P. Ashok Babu [10] performed the visualization of bubble nucleation during nucleate pool boiling outside a vertical cylindrical heated surface for ammonia–water binary and ammonia–water–lithium bromide ternary mixture.

Verla et al [11] carried out a thermodynamic design of a 2-ton solar refrigerator to preserve sea products based on aqua-ammonia intermittent vapor absorption system. The system used evacuated solar collector to energize the generator and store liquid ammonia for 18-hour operation. Their results showed that after 23 years conventional refrigeration becomes more expensive than solar refrigeration. Critoph [12] developed an ammonia-carbon refrigerator. Ammonia was utilized as a refrigerant and carbon was utilized as an adsorbent. Although the experimental setup was done producing ice, yet it was found that the coefficient of performance of the ammonia carbon refrigerator was even below 0.1. It was found that the results were unsatisfactory because of higher ammonia generation energy that was required in the generator.

Saghiruddin and Siddiqui [13] performed an economic analysis of a flat plate solar collector for operating a two stage dual fluid absorption cycle to optimize the generator temperatures. This cycle used a LiBr-H20 solution in the first stage while ammonia refrigerant with water, lithium nitrate and sodium thio-cyanate was used in the cycle at the second stage. It was found that the dual fluid-cycle using ammonia water at the second stage when operated by the flat plate solar collector shows very good performance and becomes quite economical, especially at low evaporation and high condensing temperatures. Albertson [14] presented a patent regarding solar-powered Absorption Refrigeration cycle. The system utilized LiBr-Water as the absorbent-refrigerant solution. Since LiBr-Water solution, do not require rectifier and analyzer, this presents an advantage for the system along with the fact that its operation is simple and less expensive. However, it also presents some limitations such as LiBr-Water solution cannot

operate below freezing point. Although the C.O.P for the system is high compared to NH3-Water system yet it operates at very low pressures hence efficient sealing is required which actually adds to the cost of the system.

Medrano et al [15] recommended double-lift absorption cycles working with ammonia-water refrigeration applications, which required cold at 0°C. The potentials of organic mixtures such as triflouroethanol (TFE) and tetraethyleneglycol dimethylether (TEGDME or E181) and methanol-TEGDME as working pairs in series flow and vapor exchange double-lift absorption cycles were discussed. The ammonia-water mixture was used for comparison purposes. It was found that the coefficient of performance of the vapor exchange cycle working with TFE-TEGDME is 15% higher than with ammonia-water. Pilatowsky et al [16] performed a theoretical analysis of the coefficient of performance to examine the efficiency characteristics of the monomethylamine-water solutions for a single-stage absorption system, using low generator temperatures (60-80°C), which allows the use of flat plate solar collectors. It was found that the mixture can operate at a minimum evaporator temperature of -10 degree C. The C.O.P of the mixture was found to be less than 0.3 at low evaporator temperatures, which is less compared to Aqua-ammonia mixtures.

Rivera and Rivera [17] presented a theoretical analysis of an intermittent absorption refrigeration system operating with ammonia-lithium nitrate mixture. They utilized a compound parabolic concentrator with glass-cover to operate as the generator-absorber of the cooling system. They showed that 11.8 Kg of ice could be produced at a generator

temperature of 120 degree C and a condensation temperature of 40 deg C. The C.O.P of the system was found to be between 0.15 and 0.35. Rasul and Murphy [18] performed an experimental analysis of NH3-$CaCl_2$ solutions for intermittent solar absorption refrigeration. They used parabolic solar trough as a source of generating heat. The C.O.P of the system was found to be 0.26 because ammonia is found to have less affinity towards CaCl2 compared to water. Thus higher absorption area is required which resulted in low C.O.P. for the system.

El-Shaarawi and Ramadan [19] manufactured and tested an experimental intermittent solar refrigerator using aqua-ammonia in the Egyptian climate. The C.O.P. of their system was determined to be 0.51, which is quite satisfactory. However, it was found difficult to prevent water from being transferred into condenser. It was suggested that the rectifier needs to be re-designed for efficient performance in refrigeration applications. El-Shaarawi and Ramadan [20] also investigated the effect of varying the condensing temperature on the performance of an intermittent solar refrigerator using water-ammonia solutions. It was found that for certain specified initial temperature and solution concentration and a given maximum generator temperature, decreasing the condensing temperature causes an increase in the COP of the cycle. However, for every condensing temperature, there is an optimum maximum generator temperature beyond which the COP decreases gradually. El-Shaarawi and Ramadan [21-22] also showed that the temperature of the condensate at the beginning of the absorption process has considerable influence on the performance of intermittent solar refrigerators utilizing an ammonia-water solution as the working fluid. Erickson [23] presented a patent regarding

intermittent solar-powered Absorption Refrigeration cycle. The system utilized Aqua-Ammonia as the absorbent-refrigerant solution. Compound Parabolic Concentrators (CPCs) were used as generator-absorber. Since this system did not utilize rectifier so it cannot operate below freezing point. The system utilized part of the liquid refrigerant producing the cold and part of it was utilized in cooling absorber under thermo siphon effect.

De Francisco et al. [24] developed and tested a prototype of a water-ammonia absorption system designed for solar-powered refrigeration in small rural areas. The objective of the prototype was to design a 2 kW refrigeration equipment for isolated areas with a high level of solar radiation to meet refrigeration requirements. The equipment was designed to operate with a concentrating solar power system to obtain the required temperatures for continuous operation. C.O.P. of the system was determined to be 0.05, which is quite unsatisfactory. Kim et al [25] designed a system, which is quick and light so that it can take full advantage of the energy coming from the sun. Trombe and Foex [26] reported preliminary experiments with a pilot plant using the ammonia water system. Williams el al**.** [27, 28] described a comprehensive investigation, which included a theoretical estimate of the performance of four different binary systems (methanol-silica gel, acetone-silica gel, Ammonia-water and Freon 21-tetraethylene glycol dimethyl ether and an experimental study of two of these systems (ammonia-water and freon 21 - tetraethylene glycol dimethyl ether). In both investigations, the heat source was solar radiation focused onto the generator by means of a reflector. Eisenstadt et al**.** [29] conducted tests on an experimental intermittent refrigerator employing the ammonia-

water system with solution concentrations varying from 0.4 to 0.7 and with maximum generator temperatures from 333 to 355 K. These tests have demonstrated the feasibility of the use of this cycle for air-conditioning.

Chinnappa [30] made a systematic experimental study of the intermittent vapor absorption refrigeration cycle employing two binary systems (NH3- H20 and NH3-LiNO3) with maximum solution temperatures up to about 405 K. Using solutions with an initial concentration of about 0.48, the effective cooling below 273.1 K per pound of solution, and the actual COP have been obtained, and compared with theoretical values. A solar air conditioner was built by Duffie et al**.** [31] Using lithium bromide-water as the working fluid. It was used to cool part of the Solar Energy Laboratory of the University of Wisconsin during the summer of 1962. A small and simple refrigerating apparatus using a NH3-H20 solution was built by de Sa of Pakistan [32]. The focus of 1.37 m2 parabolic reflector was used to generate ammonia vapor. Tests showed that 0.25 kg of ammonia could be extracted from 2 kg of working solution having a concentration of 53%. A simple, intermittent refrigeration system incorporating the generator-absorber with a 1.4 m2 flat plate collector was built at the University of Western Ontario [33]. NH3-H20 solutions of concentration varying from 58 to 70% were tested. Another study at the University of Western Ontario was made in 1970 [34]. This study investigated an ammonia-sodium thiocyanate solution in the same system described above. The system was unable to make any considerable amount of ice.

Sumathy et al [35] constructed an integrated solar cooling and heating system with two-stage absorption chiller (cooling capacity 100 kW). Compared to the conventional cooling system (with single stage chiller), the system with a two-stage chiller could achieve roughly the same total COP as of the conventional system with a cost reduction of about 50%. G. A. Florides, et al [36] presented a modeling and simulation of an absorption solar cooling system. The final optimized system consisted of a 15-m2 compound parabolic collector tilted 30° from the horizontal and 600 liters of hot water storage tank. Antonio et al. [37] carried out a thermodynamic simulation of a solar absorption refrigeration cycle to investigate the effect that the generator temperature and the heat exchanger efficiency make over the Coefficient of Performance (COP) and mass flux on a single absorption refrigeration system that uses solar energy as a primary source. It was found that for a constant efficiency at the heat exchanger, there is an optimum temperature to be used at the generator, while a higher temperature will decrease the system COP. Shwarts and Shitzer [38] analyzed thermodynamically the possibility to operate the solar absorption refrigeration system for air conditioning. Their results showed that the system was suitable for domestic use.

Sun [39] analyzed and performed an optimization of the water – ammonia cycle. A mathematical model was obtained that allowed the simulation of the process. Sun [40] presented a thermodynamic design and performed an optimization of the absorption refrigeration process in order to map the most common cycles for water - ammonia, and lithium bromide - water. The results can be used to select the operation conditions in order to obtain a maximum performance from the system. Sun [41] performed a

thermodynamic analysis of different binary mixtures considered in the absorption refrigeration cycle. Research activities by Brendel et al [42] in the Institute of Thermodynamics and Thermal Engineering (ITW) of the University of Stuttgart aimed at the development of a solar driven ammonia absorption chiller with a cooling power of 10 kW, together with a concept for the acclimatization of small residential and commercial buildings. With the exception of the generator, all other heat exchangers were designed as plate heat exchangers. Ammonia water was used as refrigerant absorbent solution.

The following can be concluded from the literature review:

NH3-Water presents the most suitable and reliable refrigerant-absorbent for vapor absorption systems.

Several theoretical and experimental works have been carried out on Aqua-Ammonia Intermittent solar absorption refrigeration systems

No successful work has been reported on continuously operating Aqua-Ammonia solar absorption refrigeration systems.

2.2 Main Objectives

The overall objective of the thesis is to develop a Continuously-Operated Totally Solar-Powered Refrigeration System. The specific objectives include:

- Development of new alternative designs
- Thermodynamic analysis of alternative designs
- Selection of the most suitable design

Methodology

Several alternative designs are developed and analyzed thermodynamically. To achieve the objectives, the following tasks are carried out:

- **Development of different designs**

The first task performed is the development of different designs for continuously operated solar powered refrigeration system. The new alternative designs incorporated all the possible design arrangements that may facilitate the solar powered system under consideration.

- **Performing thermodynamic analysis for different designs**

The second task performed is the thermodynamic analysis for all the designs. This provided a conceptual formulation of each alternative design, helped in defining the controlling parameters that affect the performance of each alternative design and finally determined the efficiency of each design.

- **Selection of most suitable alternative design**

Finally, the selection of best suitable design is performed that also best suits the local and regional weather conditions. This resulted in technical evaluation by parametric thermodynamic analysis to select the most suitable alternative design.

CHAPTER 3

DEVELOPMENT OF ALTERNATIVE DESIGNS

Solar energy has a great potential renewable content that can be effectively utilized for refrigeration and air-conditioning purposes using aqua-ammonia vapor absorption system. However, the biggest drawback in utilizing solar energy for continuous operation is its unavailability during the nighttime. Since solar energy is available only during the daytime so in order to meet the continuous cooling needs from the solar energy, it is needed to be integrated with a storage system that will compensate for the cooling needs at the nighttime. A categorization of solar powered cooling systems will be introduced based on different types of storage systems that can be integrated with the absorption system to meet the continuous cooling requirements. Fig 3.1 shows this categorization for solar powered aqua-ammonia vapor absorption systems. Based on different types of storage systems, three alternative designs for Solar powered Aqua-Ammonia Vapor Absorption Refrigeration and Air-Conditioning Systems are broadly introduced and categorized as follows:-

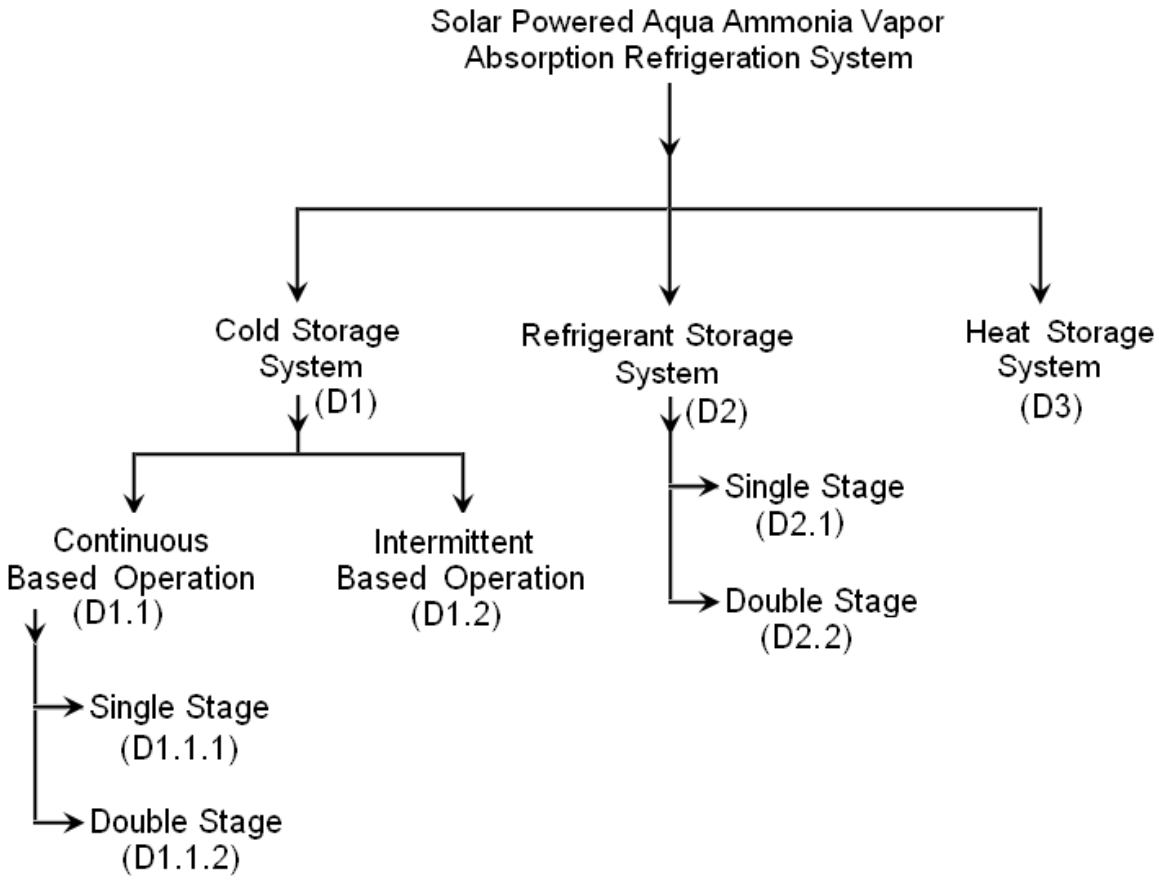

Figure 3 . 1 Classification of Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System

- Alternative Design D1 that relies on the storage of cold production by the system.
- Alternative Design D2 that relies on the storage of refrigerant generated during the available sun hours
- Alternative Design D3 that relies on storing solar thermal energy during sunrise periods for use during night.

The Cold Storage System (D1) refers to a system that produces extra refrigeration effect and stores it for use when needed to ensure a continuous cooling effect. Hence this system utilizes an additional cold storage unit. This system can be of the continuous based operation (D1.1) or the intermittent based operation (D1.2). The Continuous based operation system can further be of the single stage type (D1.1.1) or double stage type (D1.1.2). Each of these systems has its own advantages and limitations that will be discussed in detail.

The Refrigerant Storage System (D2) refers to a system in which enough refrigerant will be generated during the day time and stored to meet the continuous cooling operation. Hence this system utilizes an additional refrigerant storage unit and can be of the single stage type (D2.1) or of the double stage type (D2.2). Each of these systems has its own advantages and limitations that will be discussed in detail.

The Heat Storage System (D3) refers to a system in which enough solar energy will be stored in the form of heat during the day time to be utilized at the night time and ensure a continuous cooling operation. Hence this system utilizes an additional heat storage unit. The operational details of these alternative designs are presented in the following section.

3.1 Alternative Design (D1 – Cold Storage System)

As stated above, the main feature of cold storage system is the storage of cold storage for continuous cooling requirements. The cold storage systems can be classified into:-

- Continuous based operation (D1.1)
- Intermittent based operation (D1.2)

3.1.1 Continuous Based Operation System – (D1.1)

The cold storage continuous based operation system utilizes a conventional aqua-ammonia vapor absorption refrigeration system that operates at the daytime when the solar energy is available and produces the required refrigeration effect for the daytime as well as storing additional cooling effect in a cold storage unit. The cold storage unit then fulfills the cooling requirement at the night time when the solar energy is not available. This absorption system remains in operation at the day time and ceases to operate during the night time. This system can be either: -

- Single Stage System
- Double Stage System

3.1.1.1 Single Stage System – (D1.1.1)

Figure 3.2 shows a single stage continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on cold storage. This system operates in a conventional manner. During the daytime, solar energy is gained by the solar collector which provides the heat to the strong aqua-ammonia solution in the generator through the

thermo siphon effect in the solar collector. As heat is provided to the generator, the strong aqua-ammonia solution becomes weak as aqua-ammonia vapors are generated and flow through the rectifying column and the Dephlegmator until the water content is removed from the vapors. This requires heat extraction at the Dephlegmator.

The pure ammonia vapors then move to the condenser where condensation takes place producing saturated liquid ammonia at the exit of condenser. The saturated liquid ammonia then passes through the Vapor Liquid Heat Exchanger (VLHE) which sub-cools the liquid ammonia before it passes through the expansion valve. After expansion, a mixture of vapor and liquid is formed at low/refrigeration temperature which then passes through the evaporator and produces the required cooling effect. The cooling effect is provided by the evaporation of the liquid fraction of ammonia in the evaporator. The saturated vapor at the exit of evaporator then passes through the VLHE and becomes superheated vapor which then enters into the absorber.

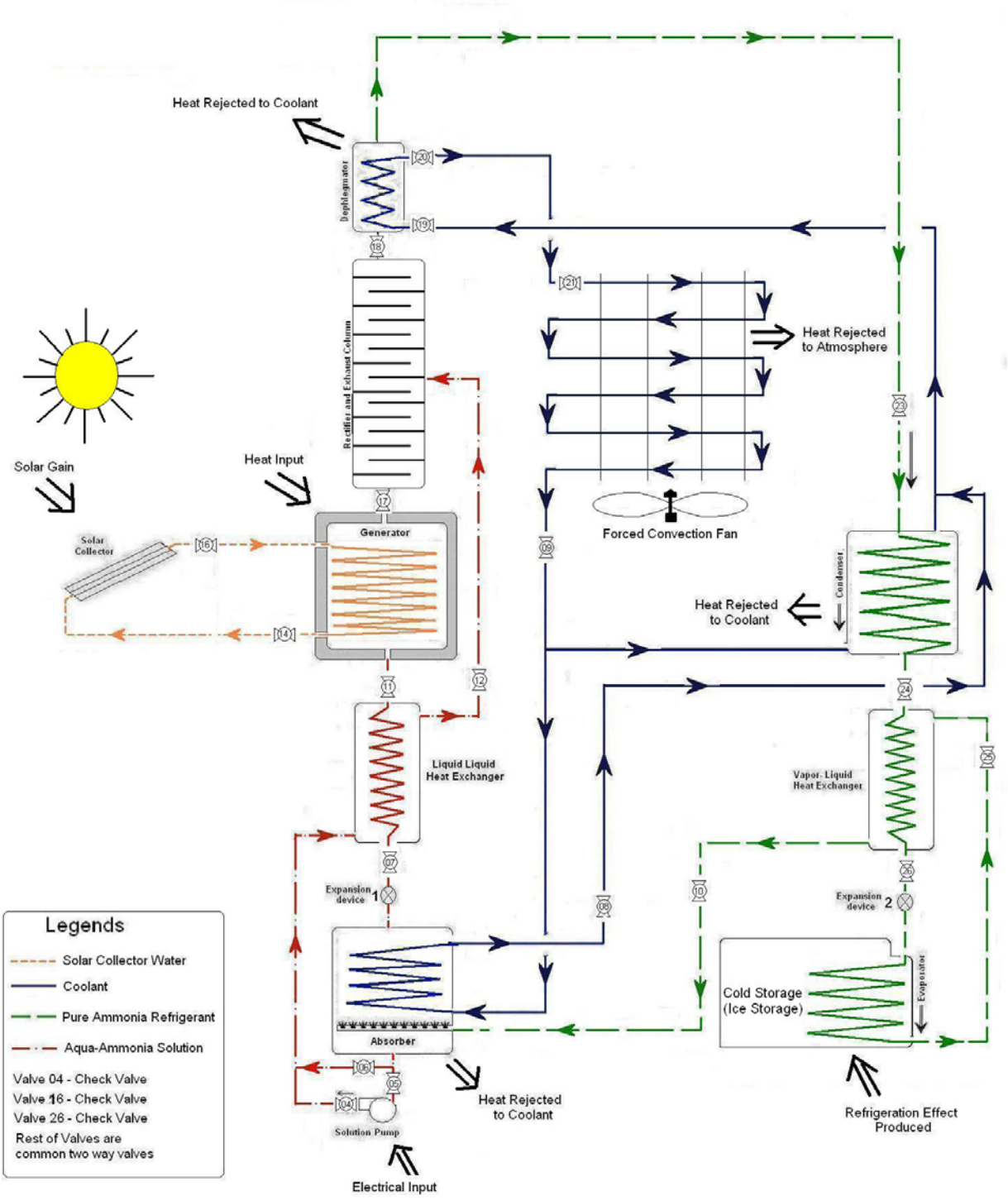

Figure 3 . 2 Single Stage Continuously Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Cold Storage

The weak aqua-ammonia solution which is at the saturation condition and at the highest temperature in the system leaves the generator and passes through the Liquid-Liquid Heat Exchanger (LLHE) where it is sub-cooled by the aqua-ammonia liquid coming from the absorber. This sub-cooled weak solution then passes through the expansion device before entering into the absorber. In the absorber the weak solution then mixes with the superheated ammonia vapor at the absorber to produce strong aqua-ammonia solution. This requires heat extraction at the absorber. The strong solution at the saturated conditions from the exit of the absorber is then pumped and pressurized through the solution pump and enters into the rectifying column then the generator after gaining heat at the LLHE.

Throughout the operation of the system, heat extraction is required from the absorber, condenser and Dephlegmator. A water cooling system designed for this purpose works on thermo siphon process. As the system is designed to reject heat to the ambient therefore, both the absorber and condenser should be designed to operate at about 3-5 ℃ higher than the ambient. Throughout the daytime operation of the system, cooling water from the coolant heat exchanger (at a temperature 3-5 ℃ lower than the condenser and absorber) initially splits into two parts i.e. one part goes to the condenser and the other part goes to the absorber, takes the heat from absorber and condenser and then mix again together to enter into the Dephlegmator. Dephlegmator operates at a temperature higher than the condenser so the cooling water then takes heat from the Dephlegmator and returns to the coolant heat exchanger for heat rejection to ambient. As the cooling water circuit is designed to operate under thermo siphon effect, therefore,

- Dephlegmator is designed to be placed at a higher elevation than that of the condenser and the absorber.
- Cooling water exit at the Dephlegmator is connected to the inlet to the coolant heat exchanger.
- Cooling water exit at the coolant heat exchanger is connected to inlet of condenser and absorber.

Coolant heat exchanger can be designed to reject heat to the ambient using natural convection or by utilizing electrical operated forced draft fan. The total electricity requirement for the operation of the whole system is based on the electrical requirement at the solution pump, at the forced draft fan (if utilized) and in the cold storage unit.

3.1.1.2 Double Stage System – (D1.1.2)

Figure 3.3 shows a double stage continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on cold storage. This system operates in a conventional manner as that explained for Single Stage System (D1.1.1). The only difference is that this system operates in two stages. The first stage operates to produce the cold water in the evaporator at a temperature lower than the ambient; however this temperature is comparatively very high to that of the evaporator temperature for single stage system. This allows solar collector of the first stage to operate at a comparatively lower temperature than that of the single stage system. The cold water produced at the evaporator of the first stage is then utilized to extract heat at the condenser, absorber and Dephlegmator of the second stage.

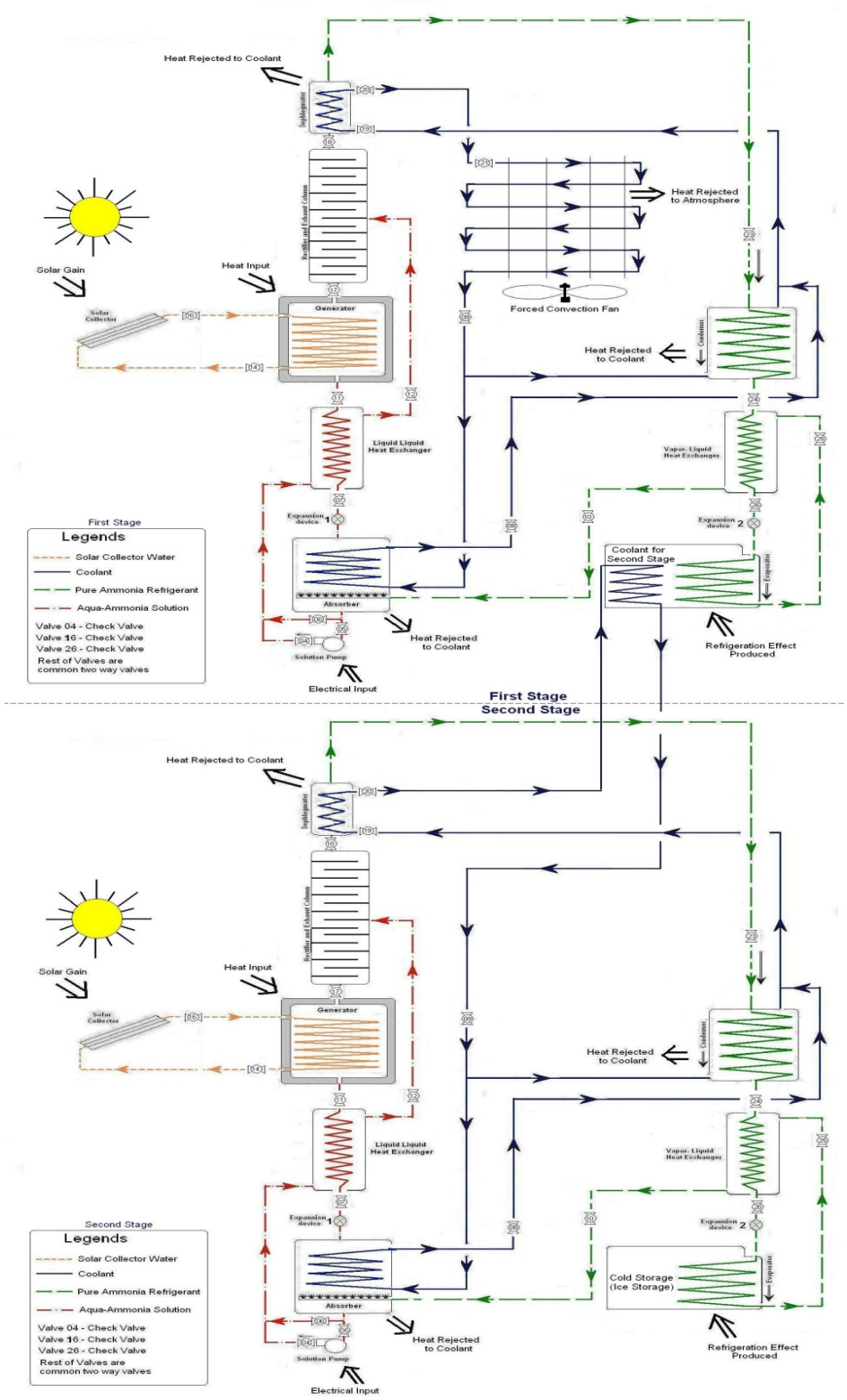

Figure 3 . 3 : Double Stage Continuously Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Cold Storage

As the condenser and absorber of the second stage can be operated at a temperature much lower than the ambient, this allows solar collector of the second stage to operate at a comparatively lower temperature than that of single stage system. The refrigeration effect is produced at the evaporator of second stage only, however the solar energy is required at the solar collector of both stages, thus the Coefficient of Performance (C.O.P.) of the double stage system would be much less compared to the C.O.P. of the single stage system. But this allows the solar collector of both the stages to operate at a much lower temperature and still produce the refrigeration effect which would not have been possible in case of single stage system. Thermo siphon effect similar to that explained earlier is used to extract heat from the absorber, condenser and Dephlegmator of both the stages, hence the condenser and absorber of the second stage must be operating at a temperature of about 3-5 ℃ higher than the evaporator temperature of first stage.

Figure 3.4 shows a typical Day and Night (24 Hour) Analysis for each component of a continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on cold storage while considering the availability of solar energy from 07:00 A.M. to 05:00 P.M. The actual operation might differ from the presented 24 hour analysis based on the actual variation of sunrise and sunset timings.

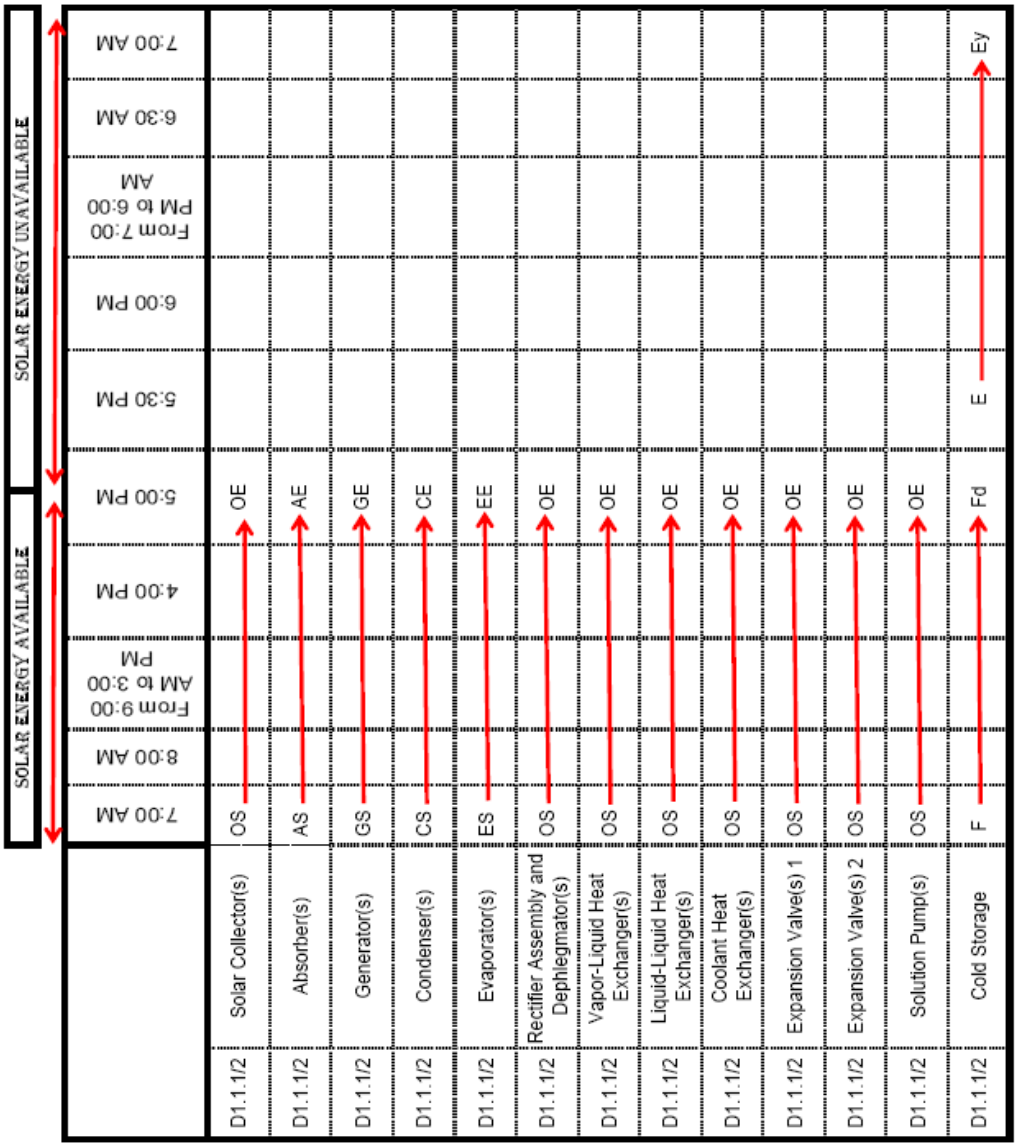

Figure 3 . 4 : 24 Hour Analysis of Single/Double Stage Continuously Operated Aqua-Ammonia Vapor Absorption Refrigeration System based on Cold Storage

AS	*Absorption Starts*	*ES*	*Evaporation Starts*	*HSS*	*Heat Storage Starts*
AE	*Absorption Ends*	*EE*	*Evaporation Ends*	*HSE*	*Heat Storage Ends*
CS	*Condensation Starts*	*F*	*Filling (Starts)*	*OS*	*Operation Starts*
CE	*Condensation Ends*	*Fd*	*Filled (Ends)*	*OE*	*Operation Ends*
GS	*Generation Starts*	*GE*	*Generation Ends*	*CLE*	*Cooling Ends*
CLS	*Cooling Starts*	*E*	*Emptying (Starts)*	*HS*	*Heating Starts*
Ey	*Empty (Ends)*	*HE*	*Heating Ends*		

Comparative Analysis of Alternative Designs D1.1

Alternatives	Advantages	Disadvantages
D1.1.1	a. Initial Cost of the system is comparatively lower. b. Comparatively less electrical energy is required for operation. c. C.O.P. of the system is comparatively higher.	a. Cost of solar collector is high. b. Cannot operate at very high ambient temperatures.
D1.1.2	a. Do not require an expensive solar collector for operation. b. Can operate at even very high ambient temperatures	a. Initial cost of the system is high because of two stages. b. Two pumps operate thus comparatively higher electrical energy requirements. c. C.O.P. of the overall system is very low.

3.1.2 Intermittent Based Operation – (D1.2)

The intermittent based operation of cold storage system utilizes an aqua-ammonia vapor absorption refrigeration system in which only generation takes place during the daytime when the solar energy is available and produce the required liquid ammonia that can be utilized to produce the cooling effect during the night time as well as storing the additional cooling effect in a cold storage unit. The cold storage unit then fulfills the cooling requirement during the day time when no the cooling effect is produced.

Figure 3.5 shows a basic intermittently operated solar powered aqua-ammonia vapor absorption refrigeration system with cold storage. During the daytime, solar energy is gained at the solar collector which provides heat to the strong aqua-ammonia solution in the generator through the thermo siphon effect in the solar collector. Initially the generator is at a low pressure. As solar heat is provided to the generator, initial pressurization takes place inside the generator and then the strong aqua-ammonia solution becomes weaker as aqua-ammonia vapors are generated and flow through the Dephlegmator until the water content from the vapors is removed. This requires heat extraction at the Dephlegmator. The condensate from the Dephlegmator moves into the Reflux Tank instead of going back into the generator. It is because the condensate from the Dephlegmator is at comparatively low temperature to that of the generator itself and thus direct entering of this condensate into the generator will adversely affect the generation process. The pure ammonia vapors from the Dephlegmator then go to the condenser where condensation takes place producing saturated liquid ammonia at the exit of condenser. This saturated liquid ammonia is stored in a Liquid Ammonia Tank throughout the day time operation of the intermittent system.

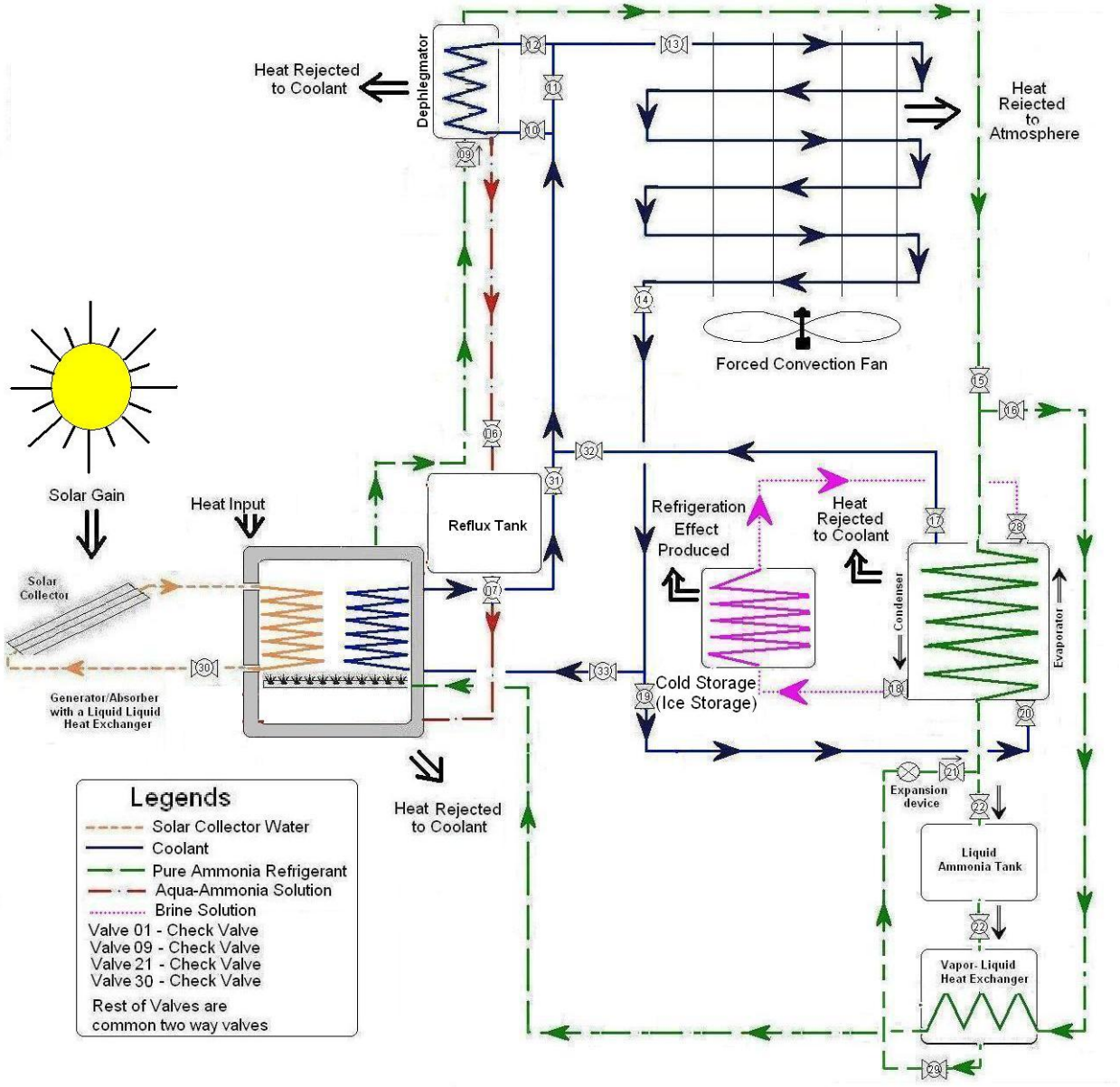

Figure 3 . 5 : Basic Intermittently Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Cold Storage

During the night, the saturated liquid ammonia then passes through the Vapor Liquid Heat Exchanger (VLHE) which sub-cools the liquid ammonia before it passes through the expansion valve. After expansion, a mixture of vapor and liquid is formed at low/refrigeration temperature which then passes through the evaporator and produces the required cooling effect.

The cooling effect is provided by the evaporation of liquid fraction of ammonia in the evaporator. The saturated vapor at the exit of evaporator passes through the VLHE and becomes superheated vapor, then enters into the absorber. The weak aqua-ammonia solution which is initially at the highest temperature and pressure in the system during the night time initially cools down so the pressure in the absorber reduces to the low pressure in the system.

In the absorber the weak solution then mixes with the superheated ammonia vapor coming from the VLHE to produce strong aqua-ammonia solution. This requires heat extraction at the absorber. Hence throughout the night time, cooling process takes place and the cooling effect is produced as well as stored in the cold storage unit for its utilization at the day time when cooling effect is not produced. After the completion of absorption process, as the day time starts, the condensate from the Dephlegmator which was stored in the Reflux Tank is entered back into the generator.

Throughout the operation of the system, heat extraction is required from the absorber, condenser and Dephlegmator. The water cooling system designed for this system works

on the thermo siphon process. As the system is designed to reject heat to the ambient therefore, the absorber is designed to operate at about 3-5 ℃ higher than the night time ambient temperature whereas the condenser is designed to operate at about 3-5 ℃ higher than the day time ambient temperature. Throughout the daytime operation of the system, cooling water from the coolant heat exchanger (at a temperature 3-5 ℃ lower than the condenser) initially moves to the condenser, takes the heat from condenser and then enters into the Dephlegmator.

Dephlegmator operates at a temperature higher than the condenser so the cooling water then takes heat from the Dephlegmator and returns to the coolant heat exchanger for heat rejection to ambient. Similarly, throughout the night time operation of the system, cooling water from the coolant heat exchanger (at a temperature 3-5 ℃ lower than the absorber) initially moves to the absorber, takes the heat from absorber and then returns to the coolant heat exchanger for heat rejection to ambient. As the cooling water circuit is designed to operate under thermo siphon effect, therefore,

- Dephlegmator is designed to be placed at a higher elevation to that of the condenser.
- Cooling water exit at the Dephlegmator is connected to the inlet to the coolant heat exchanger.
- Cooling water exit at the coolant heat exchanger is connected to inlet of condenser and absorber.

Coolant heat exchanger can be designed to reject heat to the ambient using natural convection or by utilizing electrical operated forced draft fan. The total electricity requirement for the operation of this system is based on the electrical requirement at the forced draft fan (if utilized) and in the cold storage unit.

Figure 3.6 shows a typical Day and Night (24 Hour) Analysis for each component of a basic intermittently operated solar powered aqua-ammonia vapor absorption refrigeration system based on cold storage while considering the availability of solar energy from 07:00 A.M. to 05:00 P.M. The actual operation might differ from the presented 24 hour analysis based on the actual variation of sunrise and sunset timings.

The C.O.P. of basic intermittently operated solar powered aqua-ammonia vapor absorption refrigeration system based on cold storage will be quite low because neither the LLHE nor is the rectifier used in the operation of the system. LLHE and rectifier are the components that extensively increase the performance of the system.

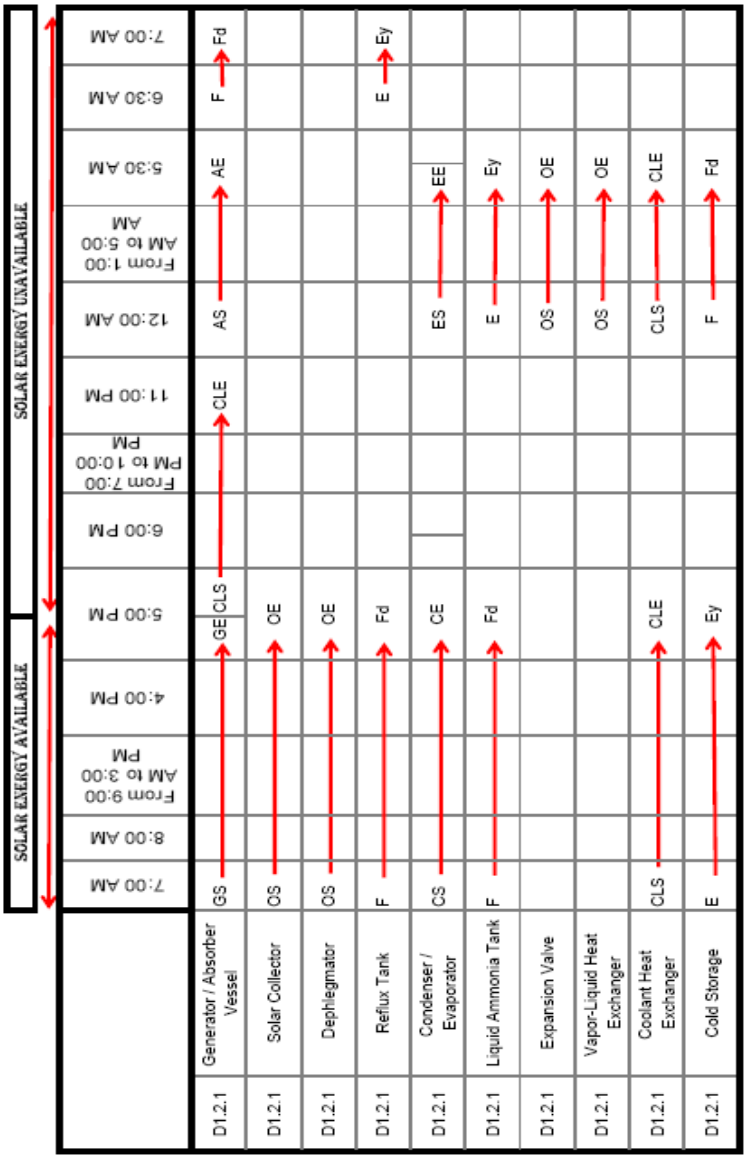

Figure 3 . 6 : 24 Hour Analysis of Basic Intermittently Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Cold Storage

AS	*Absorption Starts*	*ES*	*Evaporation Starts*	*HSS*	*Heat Storage Starts*
AE	*Absorption Ends*	*EE*	*Evaporation Ends*	*HSE*	*Heat Storage Ends*
CS	*Condensation Starts*	*F*	*Filling (Starts)*	*OS*	*Operation Starts*
CE	*Condensation Ends*	*Fd*	*Filled (Ends)*	*OE*	*Operation Ends*
GS	*Generation Starts*	*GE*	*Generation Ends*	*CLE*	*Cooling Ends*
CLS	*Cooling Starts*	*E*	*Emptying (Starts)*	*HS*	*Heating Starts*
Ey	*Empty (Ends)*	*HE*	*Heating Ends*		

3.2 Alternative Design (D2 – Refrigerant Storage System)

The main feature of the refrigerant storage systems is the utilization of a refrigerant storage unit (i.e. Liquid Ammonia Tank) for continuous cooling requirements. The continuous based operation of refrigerant storage system remains in operation day and night by utilizing a conventional aqua-ammonia vapor absorption refrigeration system to operate at the daytime when the solar energy is available and produce the required refrigerant i.e. Liquid Ammonia for the daytime as well as store the additional refrigerant in a Liquid Ammonia Tank. The liquid ammonia tank then continues the absorption process at the night time when the solar energy is not available. This system can be further classified into: -

- Single Stage System (D2.1)
- Double Stage System (D2.2)

3.2.1 Single Stage System – (D2.1)

Figure 3.7 shows a single stage continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on refrigerant storage. This system operates in a conventional manner. During the daytime, solar energy is gained by the solar collector, which provides heat to the strong aqua-ammonia solution in the generator through the thermo siphon effect in the solar collector. As heat is provided to the generator, strong aqua-ammonia solution becomes weak and the aqua-ammonia vapors generated flow through the rectifying column and the Dephlegmator until the water

content is removed from the vapors. This requires heat extraction at the Dephlegmator. The pure ammonia vapors then move to the condenser where condensation takes place producing saturated liquid ammonia at the exit of condenser.

The saturated liquid ammonia is then split into two parts i.e. one part of saturated liquid ammonia moves into Liquid Ammonia Tank and is stored for its utilization at the night time when the solar energy is not available while the other part of saturated liquid ammonia passes through the Vapor Liquid Heat Exchanger (VLHE) which sub-cools the liquid ammonia before it passes through the expansion valve. After expansion, a mixture vapor and liquid is formed at low/refrigeration temperature which then passes through the evaporator and produces the required cooling effect. The cooling effect is provided by the evaporation of liquid fraction of ammonia in the evaporator. The saturated vapor at the exit of evaporator then passes through the VLHE and becomes superheated vapor which then enters into the absorber.

The weak aqua-ammonia solution which is at the saturation condition and at the highest temperature in the system leaves the generator and passes through the Liquid-Liquid Heat Exchanger (LLHE) where it is sub-cooled for providing the heat recovery to the system. This sub-cooled weak solution then split into two parts i.e. one part moves into Weak Solution Tank and is stored for its utilization at the night time when the solar energy is not available while the other part of sub-cooled weak solution passes through the expansion device before entering into the absorber.

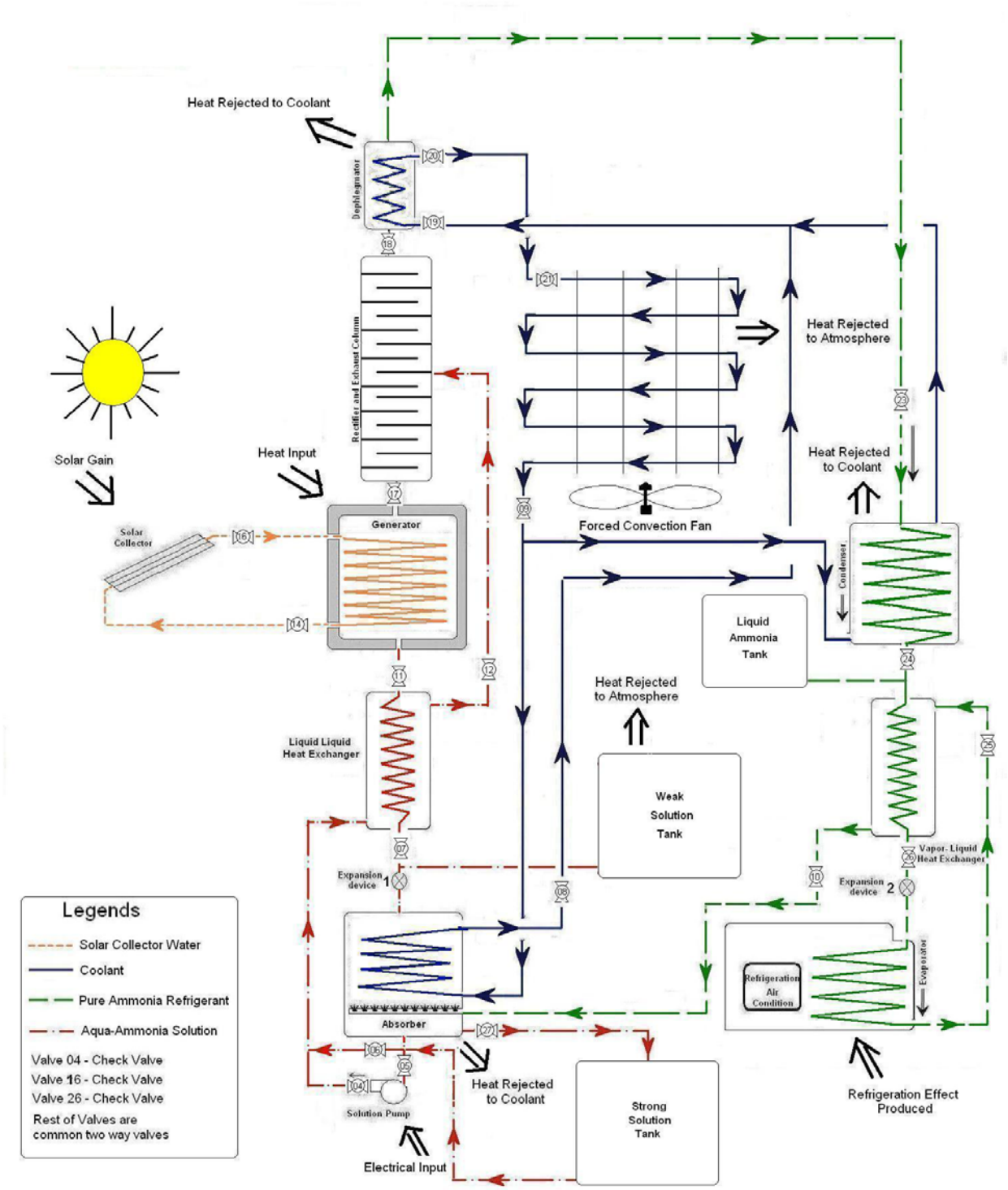

Figure 3 . 7 : Single Stage Continuously Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Refrigerant Storage

In the absorber the weak solution then mixes with the superheated ammonia vapor at the absorber to produce strong aqua-ammonia solution. This requires heat extraction at the absorber. The strong solution at the saturated conditions from the exit of the absorber is then mixed with the strong solution coming from the Strong Solution Tank. The concentration of Strong solution from Strong Solution Tank will be higher than the concentration of strong solution from the absorber because of the day and night temperature differences. This mixture results in a strong solution whose concentration is higher than the concentration of strong solution from absorber and lower than the concentration of strong solution from the Strong Solution Tank. This resulting mixture of Strong solution is then pumped and pressurized through the solution pump and enters into the generator via gaining heat at the LLHE. Throughout the day time operation of the system, liquid ammonia tank is filled with refrigerant, weak solution tank is filled with the sub-cooled weak solution and strong solution tank is being emptied from the strong aqua-ammonia solution.

As the night starts, the refrigerant stored in the liquid ammonia tank continues the absorption process. The liquid ammonia tank supplies the refrigerant for refrigeration at the evaporator which after passing the VLHE moves into the absorber. In the absorber, the superheated ammonia vapors are mixed with the sub-cooled weak solution from the Weak Solution Tank producing the strong solution which enters into the Strong Solution Tank. Thus, throughout the night time operation of the system, liquid ammonia tank is emptied from the refrigerant, weak solution tank is also emptied from the sub-cooled weak solution and strong solution tank is being filled with the strong aqua-ammonia

solution coming from the absorber. The coolant circuit for this alternative is the same as discussed in section 3.1.1.1.

3.2.2 Double Stage System (D2.2)

Figure 3.8 shows a double stage continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on refrigerant storage. This system operates in a conventional manner as that explained for Single Stage System (D2.1). The only difference is that this system operates in two stages. The first stage operates to produce the cooling water in the evaporator at a temperature much lower than the ambient; however, this temperature is comparatively very high to that of the evaporator temperature for single stage system. This allows solar collector of the first stage to operate at a comparatively low temperature to that of single stage system. This cooling water produced at the evaporator of first stage is then utilized to extract heat at the condenser, absorber and Dephlegmator of the second stage.

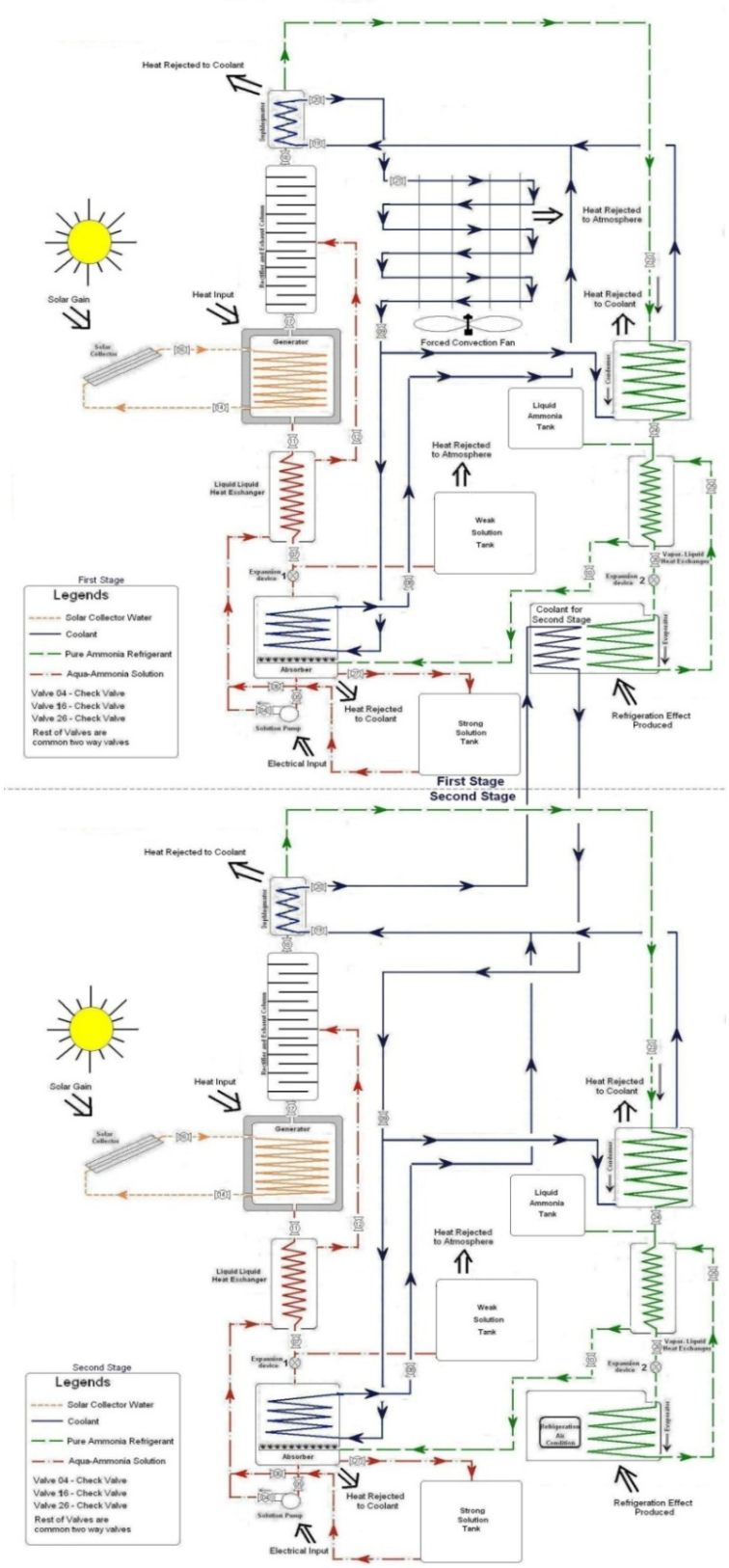

Figure 3 . 8 : Double Stage Continuously Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Refrigerant Storage

As the condenser and absorber of the second stage can be operated at a temperature much lower than the ambient, this allows solar collector of the second stage to operate at a comparatively low temperature to that of single stage system. The refrigeration effect is produced at the evaporator of second stage only, however the solar energy is required at the solar collector of both the stages, thus the Coefficient of Performance (C.O.P.) of the double stage system is much less compared to C.O.P. of the single stage system. But this allows the solar collector of both the stages to operate at a much lower temperature and still produce the refrigeration effect which would not have been possible in case of single stage system. Thermo siphon effect is used to extract heat from the absorber, condenser and Dephlegmator of both the stages, hence the condenser and absorber of the second stage must be operating at a temperature of about 3-5 ℃ higher than the evaporator temperature of first stage.

Figure 3.9 shows a typical Day and Night (24 Hour) Analysis for each component of a continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on refrigerant storage while considering the availability of solar energy from 07:00 A.M. to 05:00 P.M. The actual operation might differ based on actual variation in sunrise and sunset timings.

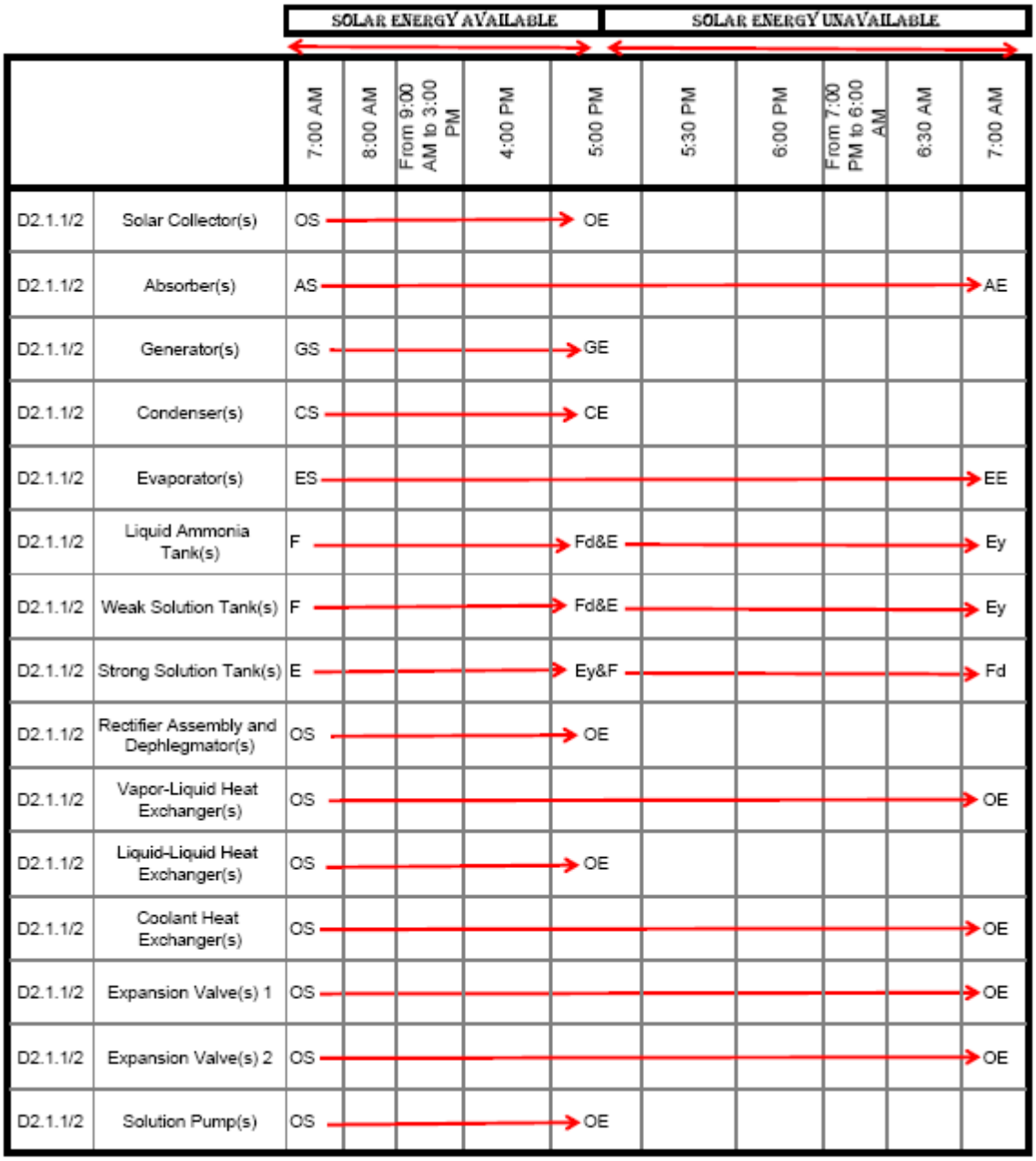

Figure 3 . 9 : 24 Hour Analysis of Single/Double Stage Continuously Operated Aqua-Ammonia Vapor Absorption System based on Refrigerant Storage

AS	*Absorption Starts*	*ES*	*Evaporation Starts*	*HSS*	*Heat Storage Starts*
AE	*Absorption Ends*	*EE*	*Evaporation Ends*	*HSE*	*Heat Storage Ends*
CS	*Condensation Starts*	*F*	*Filling (Starts)*	*OS*	*Operation Starts*
CE	*Condensation Ends*	*Fd*	*Filled (Ends)*	*OE*	*Operation Ends*
GS	*Generation Starts*	*GE*	*Generation Ends*	*CLE*	*Cooling Ends*
CLS	*Cooling Starts*	*E*	*Emptying (Starts)*	*HS*	*Heating Starts*
Ey	*Empty (Ends)*	*HE*	*Heating Ends*		

Comparative Analysis of Alternative Designs D2

Alternatives	Advantages	Disadvantages
D2.1	a. Initial Cost of the system is comparatively lower. b. Comparatively less electrical energy required for operation. c. C.O.P. of the system is comparatively higher.	a. Cost of solar collector is high. b. Cannot operate at very high ambient temperatures.
D2.2	a. Do not require an expensive solar collector for operation. b. Can operate at even very higher ambient temperatures	a. Initial cost of the system is high because of two stages. b. Two pumps operate thus comparatively higher electrical energy requirements. c. C.O.P. of the overall system is very lower.

3.3 Alternative Design (D3 – Heat Storage System)

The main feature of heat storage system is the utilization of heat storage unit for continuous cooling requirements. The continuous based operation of heat storage system utilizes a conventional aqua-ammonia vapor absorption refrigeration system to operate day and night. For such an operation, it is imperative to use a heat storage system which will store heat during the day time when the solar energy is available and utilize the stored heat at the night time when the solar energy is not available. This system remains in operation throughout the day and night time. Figure 3.10 shows a continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on heat storage. Although this system operates in a conventional manner yet it requires two sets of solar collectors.

One set of solar collector (SC1) provides solar energy to the generator for the day time operation; however the other set of solar collector (SC2) provides energy to the heat storage unit at comparatively much higher temperature to that of the generator for the night time operation. SC2 will be more expensive because it operates at much higher temperature to that of SC1. During the daytime, solar collector SC2 provides heat to the heat storage unit through thermo siphon effect and solar collector SC1 provides the heat to the strong aqua-ammonia solution in the generator through the thermo siphon effect in the solar collector. As heat is provided to the generator, strong aqua-ammonia solution becomes weak and aqua-ammonia vapors are generated which flows through the rectifying column and the Dephlegmator until the water content from the vapors is removed. This requires heat extraction at the Dephlegmator.

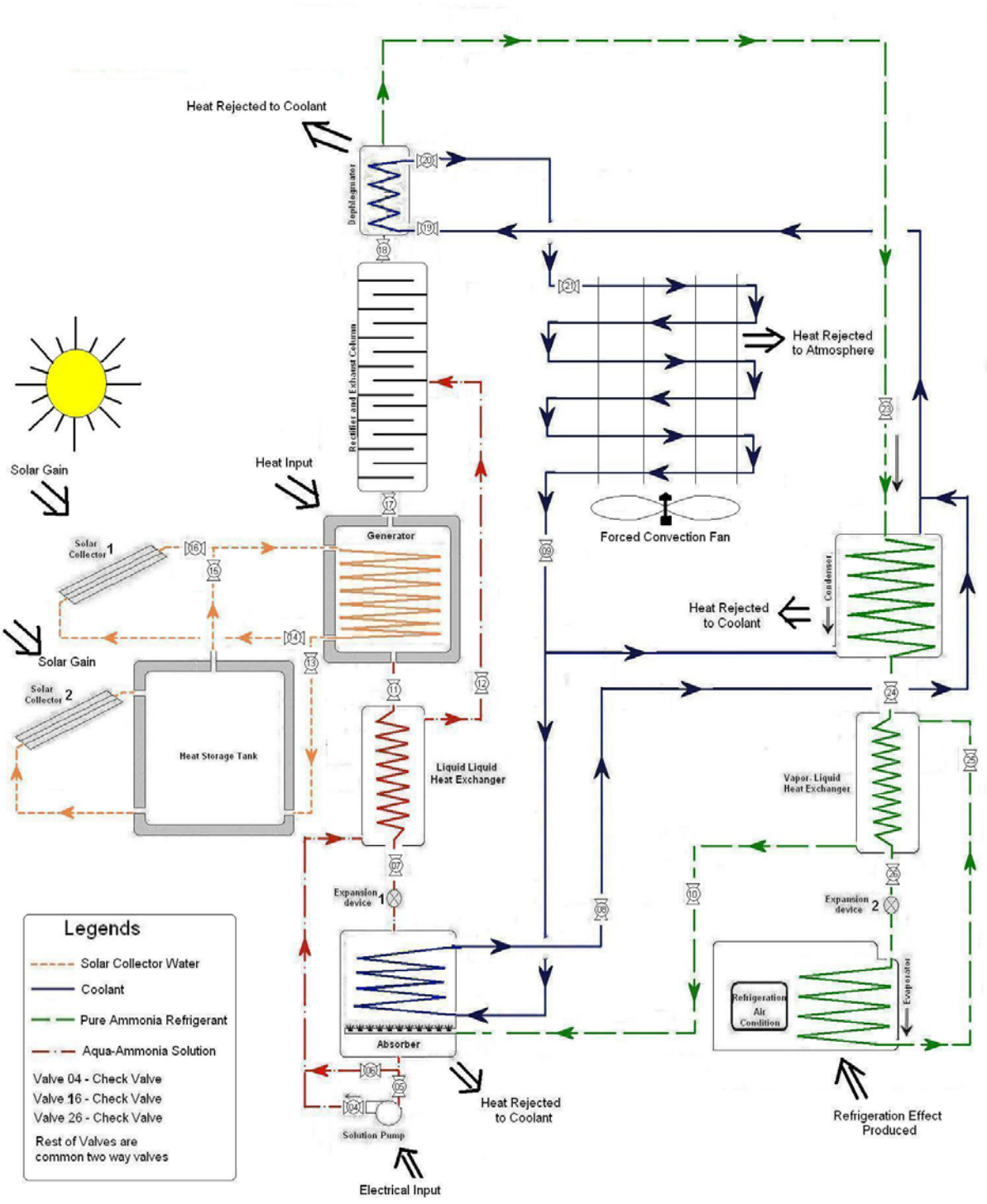

Figure 3 . 10 : Continuously Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Heat Storage

The pure ammonia vapors then move to the condenser where condensation takes place producing saturated liquid ammonia at the exit of condenser. The saturated liquid ammonia then passes through the Vapor Liquid Heat Exchanger (VLHE), which sub-cools the liquid ammonia before it passes through the expansion valve. After expansion, a mixture vapor and liquid is formed at low/refrigeration temperature, which then passes through the evaporator and produces the required cooling effect. The cooling effect is provided by the evaporation of liquid fraction of ammonia in the evaporator. The saturated vapor at the exit of evaporator then passes through the VLHE and becomes superheated vapor, which then enters into the absorber. The weak aqua-ammonia solution, which is at the saturation condition and at the highest temperature in the system, leaves the generator and passes through the Liquid-Liquid Heat Exchanger (LLHE) where it is sub-cooled for providing the heat recovery to the system. This sub-cooled weak solution then passes through the expansion device before entering into the absorber.

In the absorber, the weak solution then mixes with the superheated ammonia vapor at the absorber to produce strong aqua-ammonia solution. This requires heat extraction at the absorber. The strong solution at the saturated conditions from the exit of the absorber is then pumped and pressurized through the solution pump and enters into the generator via gaining heat at the LLHE. During the nighttime, both SC1 and SC2 cease to operate and heat storage is then directly connected to the generator to provide the required heat at the nighttime to the generator.

Throughout the operation of the system, heat extraction is required from the absorber, condenser and Dephlegmator. The water cooling system designed for this system works on the thermo siphon process. As the system is designed to reject heat to the ambient therefore, both the absorber and condenser are designed to operate at about 3-5 ℃ higher than the ambient. Throughout the operation of the system, cooling water from the coolant heat exchanger (at a temperature 3-5 ℃ lower than the condenser and absorber) initially splits into two parts i.e. one part moves to the condenser and the other part moves to the absorber, takes the heat from absorber and condenser and then again mix together to enter into the Dephlegmator. Dephlegmator operates at a temperature higher than the condenser so the cooling water then takes heat from the Dephlegmator and returns to the coolant heat exchanger for heat rejection to ambient.

Coolant heat exchanger can be designed to reject heat to the ambient using natural convection or by utilizing electrical operated forced draft fan. The total electricity requirement for the operation of this system is based on the electrical requirement at the solution pump and at the forced draft fan (if utilized). Figure 3.11 shows a typical Day and Night (24 Hour) Analysis for each component of a continuously operated solar powered aqua-ammonia vapor absorption refrigeration system based on heat storage while considering the availability of solar energy from 07:00 A.M. to 05:00 P.M. The actual operation might differ from the presented 24-hour analysis based on the actual variation of sunrise and sunset timings.

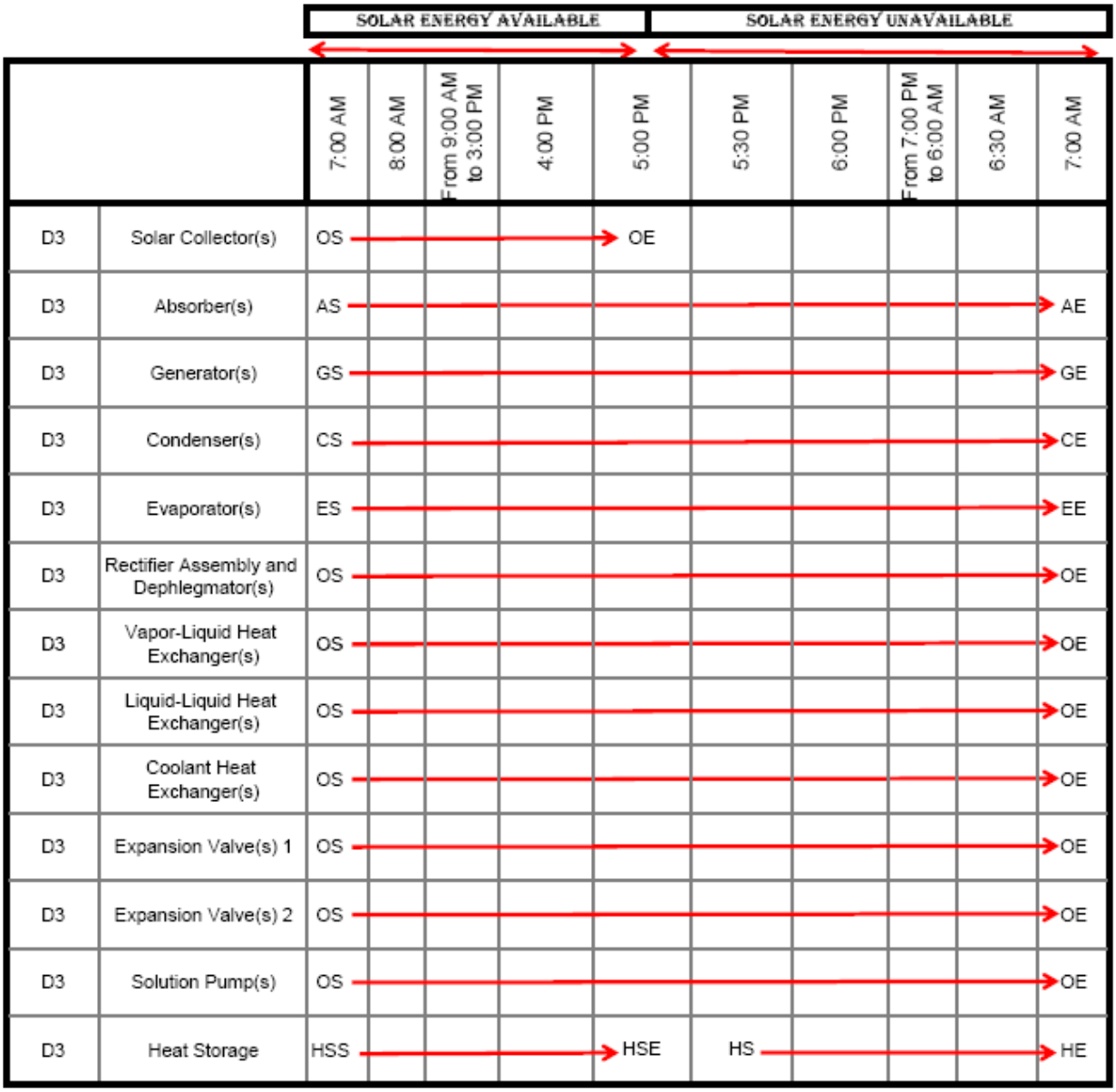

Figure 3 . 11 : 24 Hour Analysis of Continuously Operated Solar powered Aqua-Ammonia Vapor Absorption Refrigeration System based on Heat Storage

AS	*Absorption Starts*	*ES*	*Evaporation Starts*	*HSS*	*Heat Storage Starts*
AE	*Absorption Ends*	*EE*	*Evaporation Ends*	*HSE*	*Heat Storage Ends*
CS	*Condensation Starts*	*F*	*Filling (Starts)*	*OS*	*Operation Starts*
CE	*Condensation Ends*	*Fd*	*Filled (Ends)*	*OE*	*Operation Ends*
GS	*Generation Starts*	*GE*	*Generation Ends*	*CLE*	*Cooling Ends*
CLS	*Cooling Starts*	*E*	*Emptying (Starts)*	*HS*	*Heating Starts*
Ey	*Empty (Ends)*	*HE*	*Heating Ends*		

CHAPTER 4

THERMODYNAMIC ANALYSIS OF ALTERNATIVE DESIGNS

4.1 Mathematical Model of Analysis

In order to undergo, a detailed thermodynamic analysis for all the alternative designs discussed in chapter 3, it is required to undergo the mass and energy balance for each component of each alternative design. The actual refrigeration effect required to be produced at the evaporator depends upon the period over which evaporator will be operating given by Eq (4.1)

$$\text{Refrigeration}_{Effect,Day} = \text{Refrigeration}_{Effect,D\&N} \cdot \frac{24}{\text{Hours}_{Day}} \tag{4.1}$$

In case of intermittent systems, the heat duty of the evaporator is determined using the Eq (4.2) incorporating the period of absorption and depressurization in the system

$$Q_{evaporator} = \frac{\text{Refrigeration}_{Effect,Day}}{\text{Hours}_{Night} \cdot 3600 \cdot \left[\frac{E_{absorption}}{E_{absorption} + E_{Depressurization}} \right]} \tag{4.2}$$

Eq (4.3) to Eq (4.7) further defines the high and low concentrations and the pressures of the system

$$Pressure_{low} = P_{evap,out} \quad (4.3)$$

$$Conc_{high,DAY} = x_{abs,out,DAY} \quad (4.4)$$

$$Conc_{high,NIGHT} = x_{abs,out,NIGHT} \quad (4.5)$$

$$Pressure_{high} = P_{cond,out} \quad (4.6)$$

$$Conc_{low} = x_{gen,liq,out} \quad (4.7)$$

The mass flow rates of the ammonia refrigerant in the high and low pressure zones are then determined by Eq (4.8) and Eq (4.9)

$$Refrigeration_{Effect,Day} = \dot{m}_{Refrigerant,HPZ} \cdot (h_{evap,out} - h_{after,expansion}) \quad (4.8)$$

$$Refrigeration_{Effect,D\&N} = \dot{m}_{Refrigerant,LPZ} \cdot (h_{evap,out} - h_{after,expansion}) \quad (4.9)$$

The effect of vapor liquid heat exchanger is then determined using the energy balance across the vapor liquid heat exchanger as given by Eq (4.10) to Eq (4.12)

$$\square_{VLHX} = \frac{T_{vap,out,VLHX} - T_{evap,out}}{T_{cond,out} - T_{evap,out}} \quad (4.10)$$

$$h_{liq,out,VLHX} = h_{cond,out} - (h_{abs,vap,in} - h_{evap,out}) \quad (4.11)$$

$$Q_{VLHX1} = \dot{m}_{Refrigerant,LPZ} \cdot (h_{cond,out} - h_{liq,out,VLHX}) \quad (4.12)$$

In case of intermittent systems, Eq (4.13) and Eq (4.14) determine energy and heat duty of the VLHE

$$E_{VLHX1} = Mass_{Refrigerant} \cdot (h_{cond,out} - h_{liq,out,VLHX}) \quad (4.13)$$

$$Q_{VLHX} = \frac{E_{VLHX1}}{Hours_{Night} \cdot 3600 \cdot \left[\frac{E_{absorption}}{E_{absorption} + E_{Depressurization}} \right]} \quad (4.14)$$

The low and high concentrations mass flow rates are then determined using total mass balance and ammonia mass balance across the absorber under analysis as given by Eq (4.15) to Eq (4.18)

$$\dot{m}_{low,conc,abs,DAY} + \dot{m}_{Refrigerant,LPZ} = \dot{m}_{high,conc,abs,DAY} \quad (4.15)$$

$$\dot{m}_{low,conc,abs,DAY} \cdot Conc_{low} + \dot{m}_{Refrigerart,LPZ} \cdot Conc_{deph,out} = \dot{m}_{high,conc,abs,DAY} \cdot Conc_{high,DAY} \quad (4.16)$$

$$\dot{m}_{low,conc,abs,NIGHT} + \dot{m}_{Refrigerant,LPZ} = \dot{m}_{high,conc,abs,NIGHT} \quad (4.17)$$

$$\dot{m}_{low,conc,abs,NIGHT} \cdot Conc_{low} + \dot{m}_{Refrigerant,LPZ} \cdot Conc_{deph,out} = \dot{m}_{high,conc,abs,NIGHT} \cdot Conc_{high,NIGHT} \quad (4.18)$$

The mass of the refrigerant storage, weak solution and strong solution storages in case of refrigerant storage systems are determined by Eq (4.19) to Eq (4.21)

$$Storage_{Liquid,Ammonia} = (\dot{m}_{Refrigerant,HPZ} - \dot{m}_{Refrigerant,LPZ}) 3600 \cdot Hours_{Day} \quad (4.19)$$

$$Storage_{Weak,Solution} = \dot{m}_{low,conc,abs,NIGHT} \cdot 3600 \cdot Hours_{Night} \quad (4.20)$$

$$Storage_{Strong,Solution} = \dot{m}_{high,conc,abs,NIGHT} \cdot 3600 \cdot Hours_{Night} \quad (4.21)$$

The mass of the ice storage in case of ice storage systems are determined using Eq (4.22)

$$ICE_{STORAGE} = Hours_{Day} \cdot 3600 \cdot \frac{Refrigeration_{Effect,D\&N}}{\mathbf{Enthalpy}_{fusion}('Water')} \quad (4.22)$$

Similarly, the Eq (4.23) and Eq (4.24) determine the mass of heat storage for the heat storage systems

$$Q_{Heat,Storage} = Q_{generator,Night} \cdot \left[\frac{24 - Solar_{Available,Time}}{Solar_{Available,Time}}\right] \quad (4.23)$$

$$Storage_{Heat} = \frac{Q_{Heat,Storage} \cdot 3600 \cdot Solar_{Available,Time}}{\left[\mathbf{h}('Water', T = T_{gen,Night} + 40, P = Pressure) - \mathbf{h}('Water', T = T_{gen,Night}, P = Pressure)\right]} \quad (4.24)$$

Similarly, the mass flow rate going out of the strong solution storage and the mass flow rates getting into the weak solution storages are determined by Equation (4.25) to Equation (4.28).

$$\dot{m}_{high,conc,StrongSol} = \frac{Storage_{Strong,Solution}}{Hours_{Day} \cdot 3600} \quad (4.25)$$

$$\dot{m}_{high,concentration} = \dot{m}_{high,conc,abs,DAY} + \dot{m}_{high,conc,StrongSol} \quad (4.26)$$

$$\dot{m}_{low,conc,WeakSol} = \frac{Storage_{Weak,Solution}}{Hours_{Day} \cdot 3600} \quad (4.27)$$

$$\dot{m}_{low,concentration} = \dot{m}_{low,conc,abs,DAY} + \dot{m}_{low,conc,WeakSol} \quad (4.28)$$

Although in case of cold storage and heat storage systems, the mass flow rate which passes through the pump is the same as that of the mass flow rate coming out of the absorber, however in case of refrigerant storage systems, this mass flow rate is mixed with the one coming from the strong solution tank, hence the concentration and the enthalpy of the mixture is determined by the Eq (4.29) and Eq (4.30)

$$\dot{m}_{high,concentration} \cdot Conc_{pump,in} = \dot{m}_{high,conc,abs,DAY} \cdot Conc_{high,DAY} + \dot{m}_{high,conc,StrongSol} \cdot Conc_{high,NIGHT} \quad (4.29)$$

$$\dot{m}_{high,concentration} \cdot h_{pump,in1} = \dot{m}_{high,conc,abs,DAY} \cdot h_{abs,out,DAY} + \dot{m}_{high,conc,StrongSol} \cdot h_{abs,out,NIGHT} \quad (4.30)$$

Then, Eq (4.31) determines the work done by the pump

$$Q_{pump,work1} = \dot{m}_{high,concentration} \cdot (h_{pump,out} - h_{pump,in}) \quad (4.31)$$

In addition, the energy balance determines the amount of heat rejected by the absorber over the absorber as given by Eq (4.32) and Eq (4.33)

$$Q_{absorber,DAY} = \dot{m}_{low,conc,abs,DAY} \cdot h_{abs,liq,in} + \dot{m}_{Refrigerant,LPZ} \cdot h_{abs,vap,in} - \dot{m}_{high,conc,abs,DAY} \cdot h_{abs,out,DAY} \quad (4.32)$$

$$Q_{absorber,NIGHT} = \dot{m}_{low,conc,abs,NIGHT} \cdot h_{abs,liq,in} + \dot{m}_{Refrigerant,LPZ} \cdot h_{abs,vap,in} - \dot{m}_{high,conc,abs,NIGHT} \cdot h_{abs,out,NIGHT} \quad (4.33)$$

In case of intermittent systems, Eq (4.34) to Eq (4.36) determines the heat duty of the absorber

$$Total_{gen,out} \cdot h_{abs,start,ACTUAL} + Mass_{Refrigerant} \cdot h_{abs,vap,in} = E_{absorption} + (Mass_{Refrigerant} + Total_{gen,out}) \cdot h_{abs,end,ACTUAL} \quad (4.34)$$

$$E_{Depressurization} = (h_{bdp} - h_{adp}) \cdot Total_{gen,out} \quad (4.35)$$

$$Q_{absorber} = \frac{E_{absorption} + E_{Depressurization}}{Hours_{Night} \cdot 3600} \quad (4.36)$$

Similarly, the energy balance determines the amount of heat rejected by the condenser over the condenser as given by Eq (4.37)

$$Q_{condenser} = \dot{m}_{Refrigerant,HPZ} \cdot (h_{cond,in} - h_{cond,out}) \quad (4.37)$$

In case of intermittent system, the following Eq (4.38) and Eq (4.39) determine the heat rejected by the condenser

$$E_{condensation} = (h_{bc} - h_{ac}) \cdot Mass_{Refrigerant} \quad (4.38)$$

$$Q_{condenser} = \frac{E_{condensation}}{Hours_{Day} \cdot 3600 \cdot \left[\frac{E_{generation}}{E_{generation} + E_{pressurization}}\right]} \quad (4.39)$$

The rectifier effectiveness is utilized to determine the energy conservation by the rectification process, which is followed by the determination of the equation for Principle Operating Line used in determining the amount of energy required by the generation process and Dephlegmation process as given by Eq (4.40) to Eq (4.46).

$$\square_{Rectifier} = \frac{T_{gen,vap,out} - T_{deph,in1}}{T_{gen,vap,out} - T_{gen,in}} \quad (4.40)$$

$$Slope = \frac{h_{deph,in} - h_{gen,in}}{x_{deph,in} - Conc_{pump,in}} \tag{4.41}$$

$$h_{deph,in} = Slope \cdot x_{deph,in} + constant \tag{4.42}$$

$$Pole_{Generator} = Slope \cdot Conc_{low} + constant \tag{4.43}$$

$$Q_{generator} = \dot{m}_{low,concentration} \cdot (h_{gen,liq,out} - Pole_{Generator}) \tag{4.44}$$

$$Pole_{Dephlegmator} = Slope \cdot Conc_{deph,out} + constant \tag{4.45}$$

$$Q_{dephlegmator} = \dot{m}_{Refrigerant,HPZ} \cdot (Pole_{Dephlegmator} - h_{cond,in}) \tag{4.46}$$

In case of intermittent systems, a numerical analysis is performed over the differential elements of strong solution in the generator until it becomes weak. Hence, the Eq (4.47) to Eq (4.49) determines the energy of pressurization and generation

$$E_{pressurization} = (h_{ap} - h_{bp}) \cdot Mass_{Total} \tag{4.47}$$

$$Q_{generator} = \frac{E_{generation} + E_{pressurization}}{Hours_{Day} \cdot 3600} \tag{4.48}$$

$$Q_{dephlegmator} = \frac{E_{dephlegmation}}{Hours_{Day} \cdot 3600 \cdot \left[\frac{E_{generation}}{E_{generation} + E_{pressurization}}\right]} \tag{4.49}$$

Similarly, the effectiveness of Liquid-Liquid Heat Exchanger is utilized to determine the effect of LLHX over the strong and weak solutions as given by Eq (4.50) to E (4.52)

$$T_{gen,liq,in} := T_{pump,out} + \square_{LLHX} \cdot (T_{gen,liq,out} - T_{pump,out}) + 273 \tag{4.50}$$

$$Q_{LLHX} := \dot{m}_{high,concentration} \cdot (h_{gen,in} - h_{pump,out}) \tag{4.51}$$

$$h_{abs,liq,in1} := h_{gen,liq,out} - \frac{Q_{LLHX}}{\dot{m}_{low,concentration}} \tag{4.52}$$

Finally, the COP of the system is determined using Eq (4.53)

$$COP = \frac{Refrigeration_{Effect,Day}}{Q_{generator} + Q_{pump,work2}} \tag{4.53}$$

In case of intermittent systems, Eq (4.54) determines the COP

$$COP = \frac{Refrigeration_{Effect,Day}}{E_{generation} + E_{pressurization}} \tag{4.54}$$

4.2 Aqua-Ammonia Thermodynamic Properties

The knowledge of thermodynamic properties of ammonia water mixtures is essential for design, simulation and performance analysis of absorption refrigeration systems. The mixture has no environmental impact.

A number of researchers have determined the thermodynamic properties of the Ammonia water system. The vapor-liquid equilibrium (VLE) has been of primary interest due to the requirements of the absorption cycle. Park and Sonntag [43] used a generalized equation of state based on a four parameter corresponding state principle to determine the thermodynamic properties of the Ammonia water mixtures with the pressure and temperature ranges up to 200 bar and 377 deg C respectively. The extension in pressure and temperature ranges was made over the range considered by the Institute of Gas

Technology (IGT) to cover the operating range of pressure and temperature of power cycles. Ibrahim and Klein [44] used separate equations of state for liquid and gas phases for pure ammonia and pure water. They assumed the mixture to behave as an ideal solution in the liquid phase while Gibbs excess energy was used to allow a shift from ideal solution behavior in the gas phase. Their correlation covers VLE pressures and temperatures up to 110 bar and 600 K respectively. Patek and Klomfar [45] constructed correlations describing VLE properties of Ammonia water by fitting experimental data using simple functional forms. Tillner-Roth and Friend [46] presented a thermodynamic model incorporating a fundamental equation of state for the Helmholtz free energy of the Ammonia Water mixture covering the thermodynamic space between solid-liquid-vapor boundary and the critical locus. The model presented the VLE properties for pressures up to 400 bar.

Developed Correlations for Aqua Ammonia thermodynamic properties

Most of the reported Ammonia water mixture thermodynamic properties correlations used formulations with complicated mathematical structure such as Helmholtz free energy formulation and Gibbs excess energy formulation. Such formulations cannot be readily used for simulation. Therefore, the ammonia water VLE thermodynamic properties are developed in simple polynomial form that is easy to use in all applications. Hence, the experimental data (Gillespie et al. [47]) as well as data by Ibrahim and Klein [44] have been used to develop thermodynamic property correlations. The developed correlations are simple, easy to use, explicitly defined and best describes the thermodynamic properties of Ammonia water mixture.

The first correlation pertains to the saturation temperature of Ammonia water mixture in the liquid state. The saturation temperature is explicitly presented as a function of ammonia liquid mass concentration and saturation pressure of the mixture. The polynomial form of the correlation is given as:-

$$T_l = A + \sum_{i=1}^{n} (B_i x^i) + \sum_{i=1}^{n} (C_i P^i) + x \cdot \sum_{i=1}^{n-1} (D_i P^i) + x^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + x^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) \quad (4.55)$$

The second correlation pertains to the saturation temperature of Ammonia water mixture in the vapor state. The saturation temperature is explicitly presented as a function of ammonia vapor mass concentration and saturation pressure of the mixture. The polynomial form of the correlation is given as:-

$$T_v = A + \sum_{i=1}^{n} (B_i y^i) + \sum_{i=1}^{n} (C_i P^i) + y \cdot \sum_{i=1}^{n-1} (D_i P^i) + y^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + y^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) + y^4 \cdot \sum_{i=1}^{n-1} (G_i P^i) \quad (4.56)$$

The third correlation pertains to the ammonia vapor mass concentration of Ammonia water mixture. The ammonia vapor mass concentration is explicitly presented as a function of ammonia liquid mass concentration and saturation pressure of the mixture. The polynomial form of the correlation is given as:-

$$y = A + \sum_{i=1}^{n} (B_i x^i) + \sum_{i=1}^{n} (C_i P^i) + x \cdot \sum_{i=1}^{n-1} (D_i P^i) + x^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + x^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) \quad (4.57)$$

The fourth correlation pertains to the saturation enthalpy of Ammonia water mixture in the liquid state. The saturation enthalpy is explicitly presented as a function of ammonia liquid mass concentration and saturation pressure of the mixture. The polynomial form of correlation is given as:-

$$h_l = A + \sum_{i=1}^{n} (B_i x^i) + \sum_{i=1}^{n} (C_i P^i) + x \cdot \sum_{i=1}^{n-1} (D_i P^i) + x^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + x^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) \quad (4.58)$$

The fifth correlation pertains to the saturation enthalpy of Ammonia water mixture in the vapor state. The saturation enthalpy is explicitly presented as a function of ammonia vapor mass concentration and saturation pressure of the mixture. The polynomial form of the correlation is given as:-

$$h_v = A + \sum_{i=1}^{n} (B_i y^i) + \sum_{i=1}^{n} (C_i P^i) + y \cdot \sum_{i=1}^{n-1} (D_i P^i) + y^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + y^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) \quad (4.59)$$

The sixth correlation pertains to the saturation entropy of Ammonia water mixture in the liquid state. The saturation entropy is explicitly presented as a function of ammonia liquid mass concentration and saturation pressure of the mixture. The polynomial form of the correlation is given as-

$$s_l = A + \sum_{i=1}^{n} (B_i x^i) + \sum_{i=1}^{n} (C_i P^i) + x \cdot \sum_{i=1}^{n-1} (D_i P^i) + x^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + x^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) \quad (4.60)$$

The seventh correlation pertains to the saturation entropy of Ammonia water mixture in the vapor state. The saturation entropy is explicitly presented as a function of ammonia vapor mass concentration and saturation pressure of the mixture. The polynomial form of the correlation is given as:-

$$s_v = A + \sum_{i=1}^{n} (B_i y^i) + \sum_{i=1}^{n} (C_i P^i) + y \cdot \sum_{i=1}^{n-1} (D_i P^i) + y^2 \cdot \sum_{i=1}^{n-1} (E_i P^i) + y^3 \cdot \sum_{i=1}^{n-1} (F_i P^i) + y^4 \cdot \sum_{i=1}^{n-1} (G_i P^i) \quad (4.61)$$

The developed seven correlations cover the complete range of mass concentration and saturation pressure of up to 100 bar with correlation coefficients of greater than 0.99. The coefficients pertaining to the present correlations are listed in Table 4.1.

Validation of the Developed Correlations

A comparison is carried out between the Ammonia water VLE thermodynamic properties determined using current developed correlations and the ones reported in the literature. A number of these properties have been determined for different ammonia mass concentrations. The determined properties include bubble and dew point temperatures, vapor pressure, and equilibrium composition of the components, enthalpy, and entropy of the mixtures at saturation pressures. Engineering Equation Solver (EES) is used to perform calculations and generate the comparison curves. EES has inbuilt function to determine the properties of Ammonia water mixtures. This inbuilt function is based on the correlations developed by Ibrahim and Klein [44]. Hence, Ibrahim and Klein [44] correlations are implicitly used when the thermodynamic properties are directly determined by EES.

Table 4 . 1 : Coefficients for Aqua Ammonia Correlations (4.55-61)

Correlation	n	A	B_1	B_2	B_3	B_4	B_5	
1	*4*	*370.8*	*-272.3*	*93.74*	*99.21*	*-54.83*	-	
2	*5*	*365.7*	*-127.3*	*1045*	*-3733*	*5228*	*-2530*	
3	*4*	*0.1435*	*5.694*	*-12.33*	*10.53*	*-2.988*	-	
4	*4*	*405.1*	*-1758*	*-166.9*	*2598*	*-1245*	-	
5	*4*	*2651*	*-538.53167*	*-3617*	*6302*	*-3539*	-	
6	*4*	*1.261*	*-2.21*	*-5.845*	*10.58*	*-4.37*	-	
7	*5*	*7.48*	*-1.056*	*9.391*	*-39.19*	*55.02*	*-26.3*	
Correlation	**C_1**	**C_2**	**C_3**	**C_4**	**C_5**			
1	*10.11*	*-0.2768*	*0.003686299*	*-1.785E-05*	-			
2	*13.25*	*-0.5531*	*0.01268*	*-0.0001395*	*5.83E-07*			
3	*-0.01178057*	*0.0002992*	*-3.0203E-06*	*1.00E-08*	-			
4	*44.06*	*-1.195*	*0.015999011*	*-0.0000779*	-			
5	*13.58*	*-0.4555*	*0.006558325*	*-3.351E-05*	-			
6	*0.1162*	*-0.003638*	*0.00005275*	*-2.73E-07*	-			
7	*-0.1369*	*0.006572*	*-0.0001601*	*1.833E-06*	*-7.89E-09*			
Correlation	**D_1**	**D_2**	**D_3**	**D_4**	**E_1**	**E_2**	**E_3**	**E_4**
1	*-0.7369*	*0.05012*	*-0.0004538*	-	*-5.851*	*0.0470549*	*0.00007047*	-
2	*-3.711*	*0.07897612*	*0.001022*	*5.033E-06*	*9.538*	*-0.1524*	*0.001717*	*-7.935E-06*
3	*-0.01574*	*-0.0001743*	*0.00000197*	-	*0.101*	*-0.000863*	*3.258E-06*	-
4	*-4*	*0.3508*	*-0.002959*	-	*-15.94*	*-0.251*	*0.003846*	-
5	*-1.766*	*-0.01591*	*-0.00002691*	-	*13.7*	*-0.06589*	*0.0007111*	-
6	*0.04415*	*-0.000176*	*-8.65E-07*	-	*-0.03866*	*-0.000358*	*7.6646E-06*	-
7	*-0.01482*	*0.0006243*	*-8.639E-06*	*4.38E-08*	*0.07947493*	*-0.003291*	*0.00004413*	*-2.21E-07*
Correlation	**F_1**	**F_2**	**F_3**	**F_4**	**G_1**	**G_2**	**G_3**	**G_4**
1	*4.564*	*-0.06016*	*0.000141*	-	-	-	-	-
2	*-17.53*	*0.2862*	*-0.00322449*	*0.00001503*	*8.4*	*-0.1174*	*0.001222*	*-5.517E-06*
3	*-0.07766*	*0.0008663*	*-4.077E-06*	-	-	-	-	-
4	*12.76*	*0.04592*	*-0.001865*	-	-	-	-	-
5	*-14.77*	*0.1091*	*-0.0009271*	-	-	-	-	-
6	*0.008434*	*0.000355*	*-5.891E-06*	-	-	-	-	-
7	*-0.1461*	*0.006177*	*-0.0000834*	*4.21E-07*	*0.07477*	*-0.003363*	*4.5612E-05*	*-2.31E-07*

The following equations reported by Patek & Klomfar [45] are similarly used to determine the thermodynamic properties.

$$T(p,x) = T_0 \sum_i a_i (1-x)^{m_i} \left[\ln\left(\frac{p_0}{p}\right)\right]^{n_i}$$

$$T(p,y) = T_0 \sum_i a_i (1-y)^{m_i/4} \left[\ln\left(\frac{p_0}{p}\right)\right]^{n_i}$$

$$y(p,x) = 1 - \exp\left[\ln(1-x) \sum_i a_i \left(\frac{p}{p_0}\right)^{m_i} x^{n_i/3}\right]$$

$$h_l(T,x) = h_0 \sum_i a_i \left(\frac{T}{T_0} - 1\right)^{m_i} x^{n_i}$$

$$h_g(T,y) = h_0 \sum_i a_i \left(1 - \frac{T}{T_0}\right)^{m_i} (1-y)^{n_i/4}$$

The data reported in Table 3 by Park and Sonntag [43] and the data reported in Table 4 by Tillner and Friend [46] have been entered into the look up tables of EES to determine the required properties. Finally, as all the data have been entered into EES, parametric tables have been generated to compare the thermodynamic properties determined by different approaches. The results are presented in graphical forms.

The first comparative result pertains to the bubble-point (incipient boiling) temperature. Figure 4.1 shows a comparison of bubble point temperatures of Ammonia water mixtures determined using different correlations plotted against Ammonia Liquid Mass Concentration at a pressure of 50 bar (725 psia). The figure indicates that the results obtained using different correlations are in good agreement with each other and with the present work.

The bubble point temperature obtained using correlation developed by Ibrahim & Klein [44] and Tillner & Friend [46] exhibit a very close agreement with those obtained using the correlation developed by Park & Sonntag [43] and Patek & Klomfar [45]. The maximum deviation observed between the values of bubble point (incipient boiling) temperatures obtained using the present correlation and those developed by Park & Sonntag [46] are less than 3 % in the range of 0.3 to 0.5 ammonia liquid mass concentrations.

The second comparative property is the dew-point temperature. Figure 4.2 shows a comparison of dew point temperatures of Aqua-Ammonia mixtures plotted against Ammonia Vapor Mass Concentration at a pressure of 50 bar (725 psia). It is clear from the plot that all the results, with the exception of those by Tillner and Friend [46] are in good agreement with each other.

In other words, the dew point temperature by Ibrahim & Klein [44], Park & Sonntag [43] and Patek & Klomfar [45] and are in good agreement with those determined by the present correlation. However the values by Tillner & Friend [46] are slightly different from the others at low ammonia vapor mass concentrations. The average deviation between the dew point temperatures of Tillner & Friend [46] and others can be more than 3% for ammonia vapor mass concentration values less than 0.6.

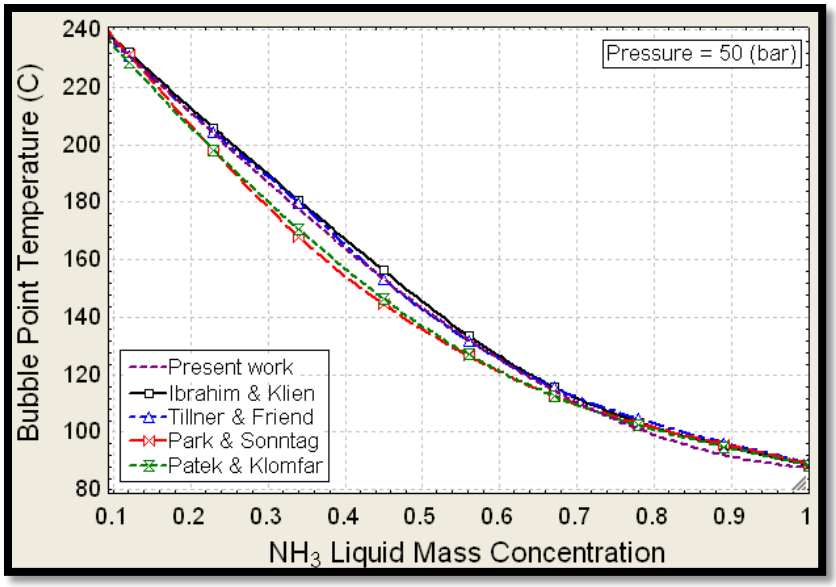

Figure 4 . 1 : Bubble Point Temperatures of Ammonia water mixture at a pressure of 50 bar (725 psia)

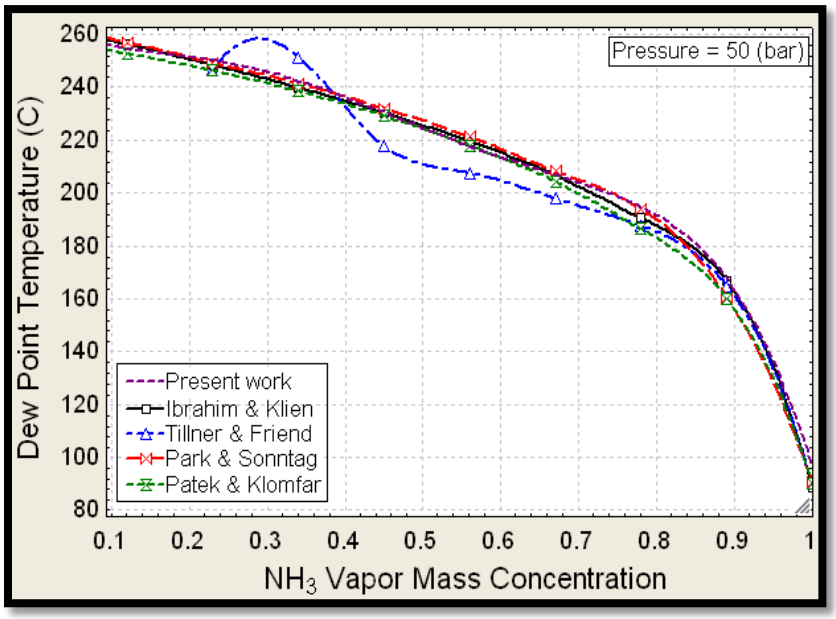

Figure 4 . 2 : Dew Point Temperatures of Ammonia water mixture at a pressure of 50 bar (725 psia)

The third comparative property is the liquid enthalpy. In thermodynamic analyses, usually the enthalpy difference is used rather than the absolute value of the enthalpy. Thus, one should be aware that values of enthalpies given in property tables and/or charts in different references might be referred to different datum (for zero enthalpy). Therefore, comparisons for enthalpies from different sources should be based on enthalpy changes rather than individual enthalpy values. Figure 4.3 shows a comparison of saturated liquid enthalpy change of Aqua-Ammonia mixtures with respect to the enthalpy at zero ammonia mass concentration plotted against Ammonia Liquid Mass Concentration at a pressure of 50 bar (725 psia). The figure indicates that the results determined using correlation by Park & Sonntag [43] shows the highest enthalpy change whereas the results determined using correlations by Patek & Klomfar [45] shows the lowest enthalpy change. The result obtained using different correlations exhibit the same trend, i.e. an increase in the enthalpy change until the point of inflection and then a decrease in the enthalpy change afterwards. The negative values in the enthalpy change signify the fact that the enthalpy at zero ammonia mass concentration is higher than the data under consideration. The enthalpy change by Ibrahim & Klein [44] is in good agreement with the results of the present study with a deviation of less than 0.7%. Other enthalpy change results show more deviations from both of Ibrahim & Klein [44] and those of the present work.

Figure 4.4 shows a comparison of saturated vapor enthalpy change of Aqua-Ammonia mixtures with respect to the enthalpy at zero ammonia mass concentration plotted against Ammonia Vapor Mass Concentration at a pressure of 50 bar (725 psia).

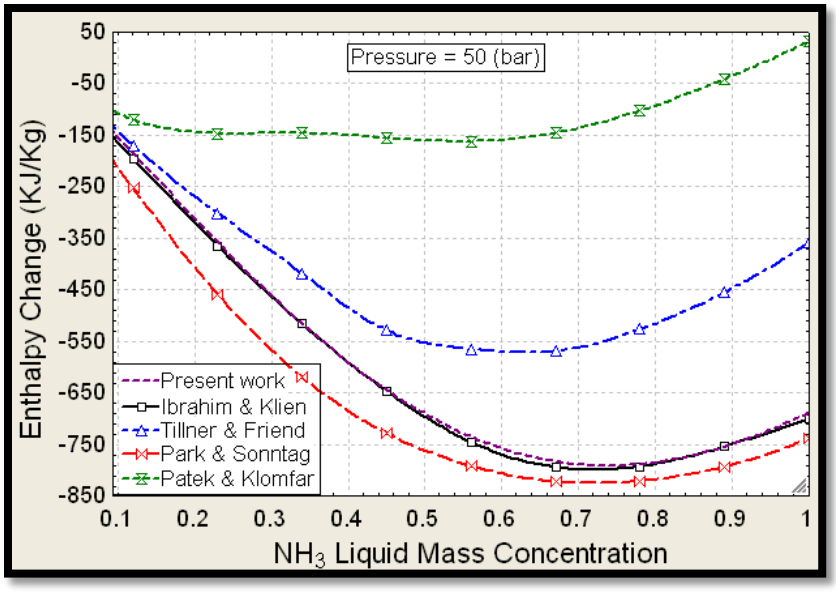

Figure 4 . 3 : Enthalpy Change for Saturated Liquid aqua-ammonia mixture at a pressure of 50 bar (725 psia)

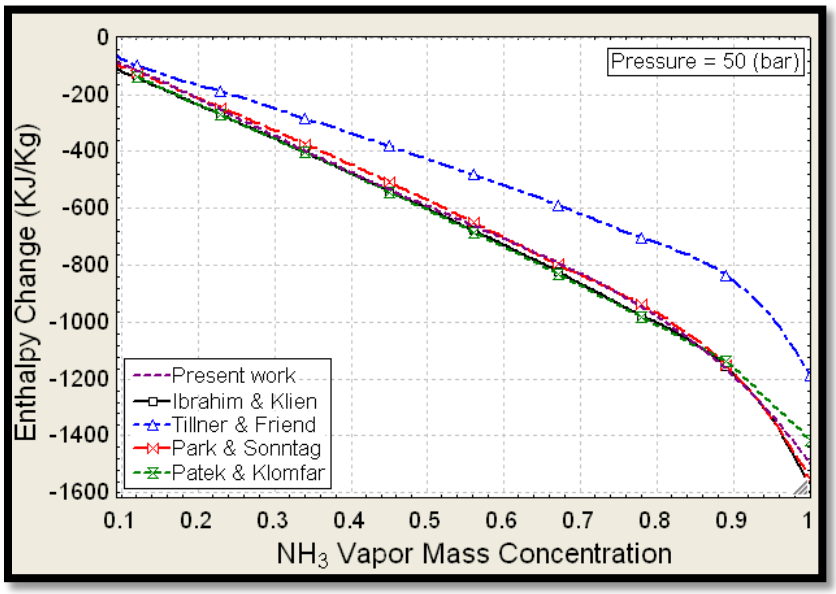

Figure 4 . 4 : Enthalpy Change for Saturated Vapor aqua-ammonia mixture at a pressure of 50 bar (725 psia)

The Figure 4.4 indicates that the results by Ibrahim & Klein [44], Park & Sonntag [43] and Patek & Klomfar [45] shows a good agreement with the results of the present study, however the values of enthalpy change determined using Tillner & Friend [46] correlation are less compared to the others' values. The average deviation between the enthalpy change values for saturated vapor mixtures determined by present study, Ibrahim & Klein [44], Park & Sonntag [43] and Patek & Klomfar [45] is less than 2%.

The fifth comparative property pertains to the saturated liquid entropy. Figure 4.5 shows a comparison of entropy change of saturated liquid Aqua-Ammonia mixtures with respect to the entropy at zero ammonia mass concentration plotted against Ammonia Liquid Mass Concentration at a pressure of 50 bar (725 psia). The figure indicates that that the results obtained using Park & Sonntag [43] correlation shows the highest entropy change values whereas the results obtained using Tillner & Friend [46] correlation shows the lowest value of entropy change. The point of inflection occurs at around 0.75 ammonia liquid mass concentrations for the properties using Ibrahim & Klein [44] and Park & Sonntag [43] however it occurs at around 0.6 ammonia liquid mass concentrations for the value obtained using Tillner & Friend [46] correlations. In general, the entropy change obtained using Ibrahim & Klein [44] is lower than Park & Sonntag [43] however at the saturation lines (i.e. at pure water and pure ammonia), the values of entropy change are identical for both. Figure 4.6 shows a comparison of entropy change of saturated vapor Aqua-Ammonia mixtures with respect to the entropy at zero ammonia mass concentration plotted against Ammonia Vapor Mass Concentration at a pressure of 50 bar (725 psia).

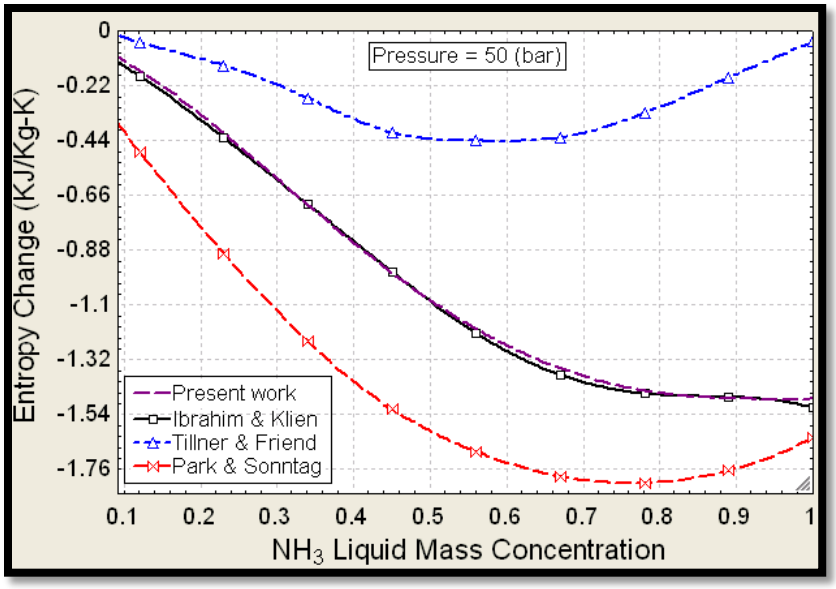

Figure 4 . 5 : Entropy Change for Saturated Liquid aqua-ammonia mixture at a pressure of 50 bar (725 psia)

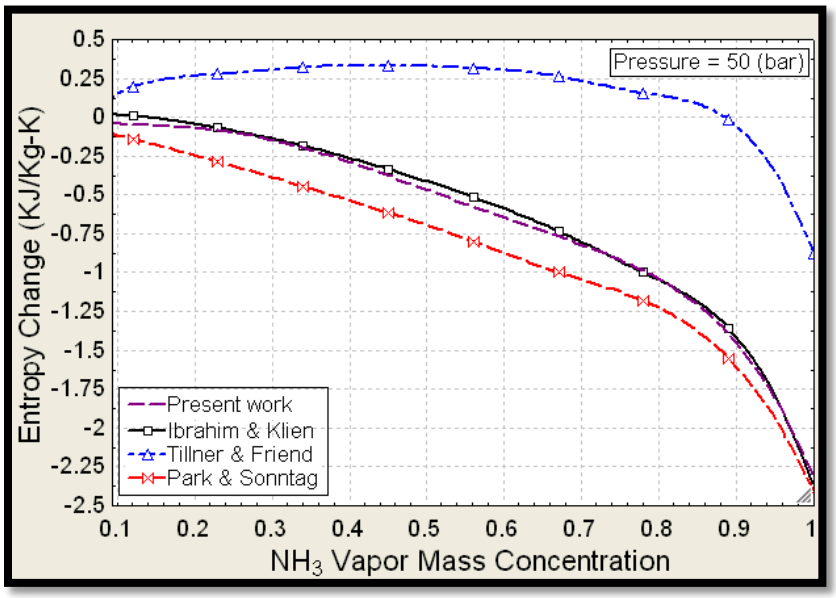

Figure 4 . 6 : Entropy Change for Saturated Vapor aqua-ammonia mixture at a pressure of 50 bar (725 psia)

The Figure 4.6 indicates that the results by Ibrahim & Klein [44] and Park & Sonntag [43] show good agreement with the present study correlation. However, the results of entropy change obtained using the correlation developed by Tillner & Friend [46] is less compared to the others values. The average deviation between the entropy change data for saturated vapor mixtures obtained by present study, Ibrahim & Klein [44] and Park & Sonntag [43] is less than 5%.

Hence, a set of seven polynomial forms of correlations that are simple, easy to use, explicitly defined and best describes the thermodynamic VLE properties of Ammonia water mixture have been developed based on experimental data. The present correlations give results that are particularly in excellent agreement with the results obtained using the correlations developed by Ibrahim and Klein [44].

4.3 Basic Performance Parameters

In order to analyze the alternative designs detailed in chapter 3, it is important to determine the basic performance parameters for analysis of such systems. The basic performance parameters are those parameters, which directly influence the performance of a system. Such parameters required for a detailed thermodynamic analysis are listed:-

- Generator/Condenser Pressure (High Pressure in the system)
- Absorber/Evaporator Pressure (Low Pressure in the system)
- Absorber Exit Concentration (Highest Aqua-Ammonia Concentration in the system)
- Generator Exit Concentration (Lowest Aqua-Ammonia Concentration in the system)
- Ammonia Purification Level (Highest Concentration of Ammonia refrigerant in the system)
- Total Refrigeration Effect Required
- Desired Cooling Load temperature
- Effectiveness of Liquid-Liquid Heat Exchanger (LLHE)
- Effectiveness of Vapor-Liquid Heat Exchanger (VLHE)
- Effectiveness of Rectifier

Having knowledge of the above mentioned basic performance parameters; the actual system under consideration can be effectively analyzed in detail.

4.4 Pressures and Concentrations Selection

The performance of continuous conventional and intermittent solar absorption refrigeration systems depends mainly on four operating parameters, namely, the two operating pressures (high/condenser pressure and low/evaporator pressure) and the highest and lowest operating ammonia mass concentrations. In designing solar-driven intermittent aqua-ammonia absorption refrigeration systems the selection of these four operating parameters needs a detailed analysis. The above four operating parameters in intermittent solar refrigeration systems are dependent on four operating temperatures, namely, the condenser (day-ambient), the absorber (night-ambient), the evaporator (cooling-load), and the maximum generator temperatures. This section clarifies the effect of the operating temperatures on the selection of the four operating parameters for both continuous and intermittent systems.

Figure 4.7 and 4.8 show typical continuous conventional and intermittent solar aqua-ammonia absorption refrigeration systems, respectively. The cooling effect is produced at the evaporator when heat is extracted from the low-temperature load by means of the low-pressure refrigerant inside the evaporator.

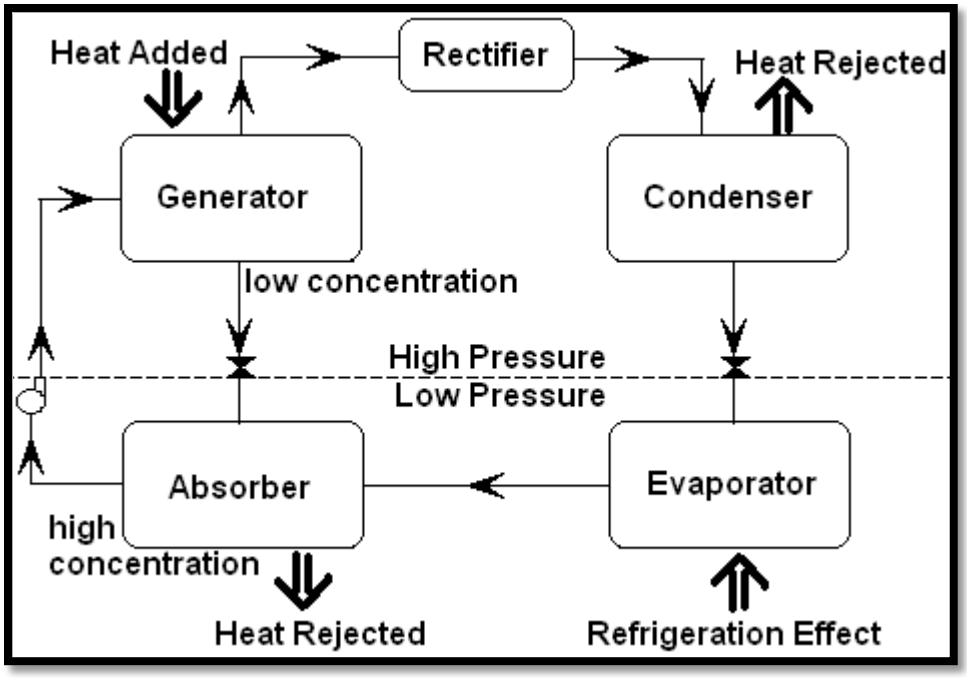

Figure 4 . 7 : Typical Continuous Aqua-ammonia Absorption Refrigeration System

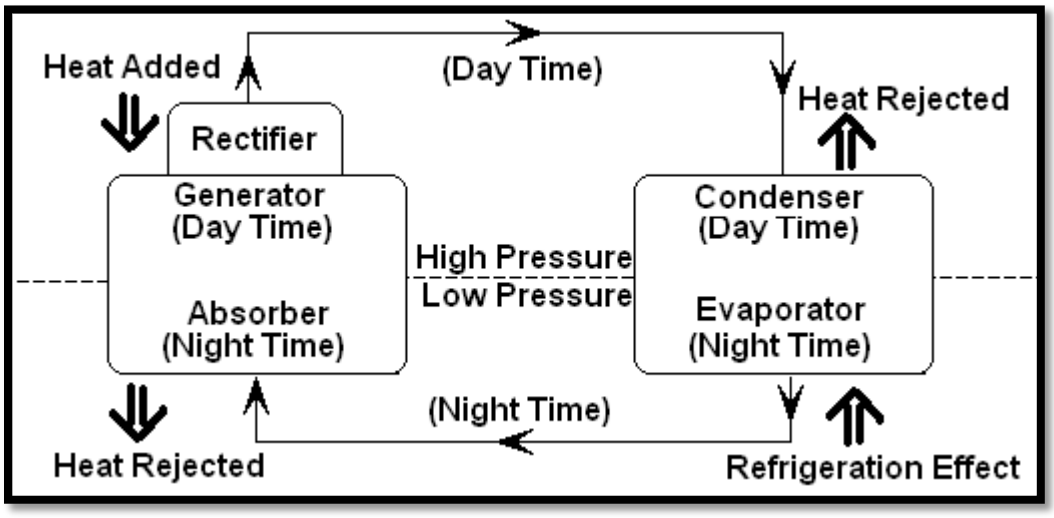

Figure 4 . 8 : Typical Intermittent Solar Aqua-ammonia Absorption Refrigeration System

The low-pressure value inside the evaporator corresponds to the saturation pressure at the refrigerant evaporation temperature, which should be a few degrees (3-5 ˚C) less than the load temperature (i.e. the temperature at which the refrigeration effect is required). Thus, the load minimum temperature defines the evaporation temperature and hence the evaporation pressure. This same low pressure equals also to the absorber pressure.

The absorption process is a non-isothermal constant-pressure heat rejection saturation process. The absorber usually rejects heat to the ambient. Therefore, the temperature at the end of the absorption process should be a few degrees (3-5 ˚C) higher than the available ambient temperature (T_0). At the end of the absorption process the aqua-ammonia concentration has its highest value in the cycle. Thus, the temperature at the end of the absorption process (T_0 + (3-5 ˚C)), the absorption/evaporator pressure (= P_{low}), and the conditions of being saturated during the absorption process, define the value of the highest aqua-ammonia concentration in the system.

Similarly, condensation of ammonia occurs in the condenser at the system's high-pressure value (P high) by rejecting heat to the ambient. The high-pressure value inside the condenser (P high) corresponds to the saturation pressure at the refrigerant condensation temperature, which should be a few degrees (3-5 ˚C) higher than the ambient temperature (T_0 + (3-5 ˚C)). Thus the ambient temperature defines the condensation temperature and hence the condenser pressure. This same high pressure equals also to the ammonia generation pressure. The ammonia generation (desorption) process is a non-isothermal constant-pressure heat addition saturation process. The generator receives heat from the available heat source (burning fuel, waste heat, solar

energy, etc). Therefore, the temperature at the end of the generation process is the maximum available temperature for the aqua-ammonia in the cycle and is dependent on the available heat source temperature. At the end of the generation process the aqua-ammonia concentration has its lowest value in the cycle. Thus, the temperature at the end of the generation process (that depends on the available heat source), the generation pressure (= P_{gen}), and the conditions of being saturated during generation, define the value of the lowest aqua-ammonia concentration in the system.

The above discussion is also applicable on Fig. 4.8, which shows the simplest possible intermittent solar absorption refrigeration system. In this system, the condenser and the generator are active during the daytime while they work during the night time as evaporator and absorber, respectively. In other words, this simple intermittent solar refrigeration system is comprised of only two units; the condenser/evaporator unit and the generator/absorber unit. In this intermittent daily cycle, the maximum temperature at the end of the generation process is dependent on the type of solar collector used to collect the available solar energy (flat plate, evacuated tube or concentrating collector) and the available solar intensity in kW/m^2. The condenser temperature is dependent on the day-time ambient temperature and the absorption temperature is dependent on the night-time ambient temperature.

In summary, the design of a vapor absorption refrigeration system is dependent upon the selection of the high and low pressures, which are dependent on the ambient and load temperatures, respectively, as well as the highest and lowest ammonia mass concentrations, which are dependent on the ambient and the maximum available heat source temperatures, respectively. For the intermittent solar refrigerator that works on a

daily basis, two ambient temperatures are important, the day-time and the night time ambient temperatures.

The results of a thermodynamic analysis that clarifies the effect of the controlling four temperatures on the selection of the four operating parameters (two pressures and two concentrations) are presented as follows.

The software: Engineering Equation Solver (EES) is utilized to produce all the presented results and plots as well as conducting the analysis of the absorption cycle processes. EES also has a built in procedure for thermodynamic properties of aqua-ammonia solutions. Thus multiple iterations of the procedure can be performed effectively for providing the right solutions to the problem under analysis.

Figure 4.9 shows the pure ammonia boiling/condensation (saturation) curve over a selected evaporator low-temperature range. Hence, Fig. 4.9 gives the evaporation temperature of pure ammonia (which should be a few degrees less than the required load temperature) against the low (evaporator/absorber) pressure (P_{low}) in the system. From Fig. 4.9, it is clear that for sub-zero (freezing) refrigeration purposes, the low pressure in the aqua-ammonia vapor absorption systems must be below 4bar. However, for air-conditioning purposes, it can be as high as 8 bar. It can also be seen from Fig. 4.9 that below 4.9 bar, the low pressure in the system has a more significant effect on the evaporation temperature as compared to the corresponding effect on the evaporation temperature at pressures higher than 2 bar. Figure 4.10 gives the highest ammonia mass concentration (at the absorber exit) against the low (evaporator/absorber) pressure in the system for various selected absorber exit temperatures (which equals to T_0 + a few degrees; from -15˚C to 75 ˚C).

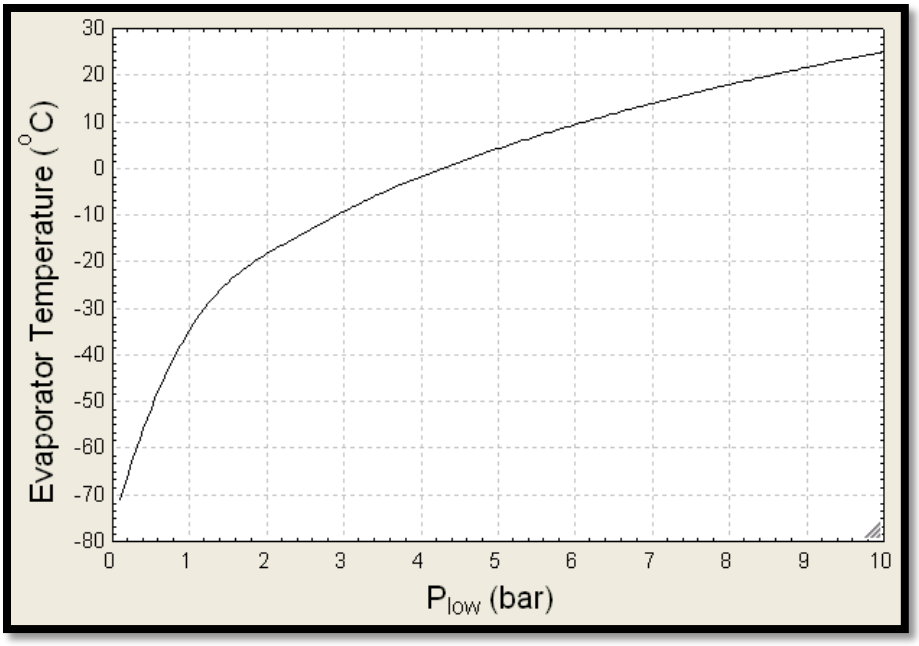

Figure 4 . 9 : Evaporator Temperature versus low pressure in the absorption system

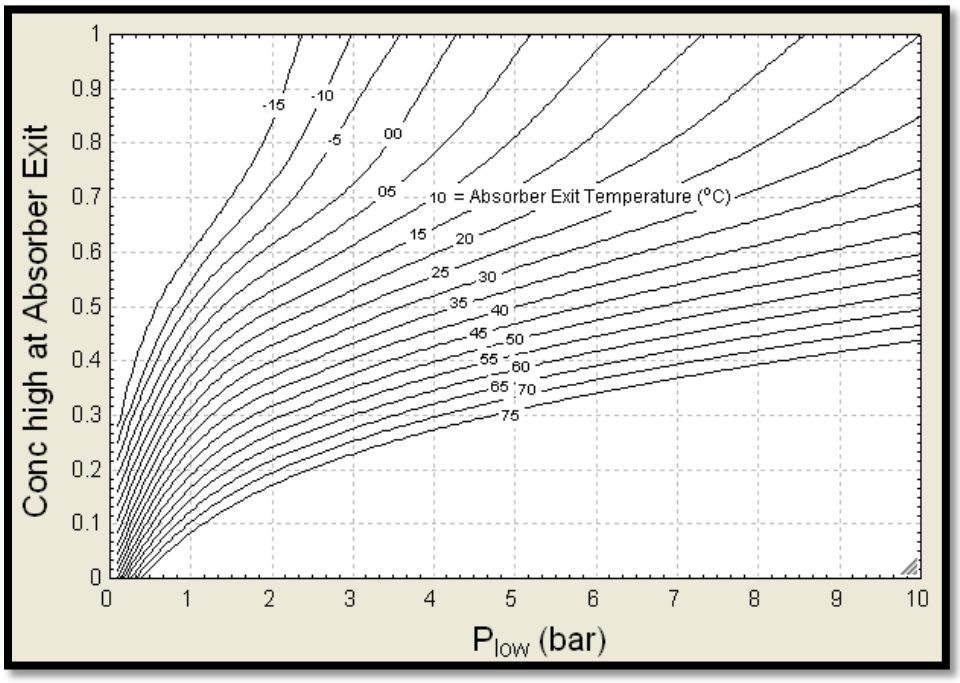

Figure 4 . 10 : Highest NH3 Concentration in aqua-ammonia (at absorber exit) v/s low (evaporator/absorber) pressure in the absorption system

From Fig. 4.10, it can be seen that, for a given absorber exit temperature, reducing the low pressure in the system (i.e. reducing the load temperature) causes a decrease in the value of the highest ammonia mass concentration (at the absorber exit or the end of the absorption process). On the other hand, it can be seen from Fig. 4.10 that, for a given Plow (evaporator/absorber pressure, i.e., load temperature) increasing the absorber exit temperature reduces also the value of the required highest concentration in the system. Combining Fig. 4.9 and Fig. 4.10, the required highest ammonia mass concentration in the system (at the absorber exit) against the evaporator temperature (load temperature minus a few degrees) can be obtained for various selected absorber exit temperatures (which equals to T_0 + a few degrees; from -15˚C to 75 ˚C) as shown in Fig. 4.11.

This reduction in the value of the highest possible refrigerant concentration in the absorption refrigeration system would negatively affect the system's performance. This is because it reduces the concentration difference (between the highest and lowest concentrations) in the system, thus it directly decreases the amount of ammonia desorbed in the generator and consequently decreases the refrigeration effect produced at the evaporator. Hence at very low evaporator pressures (low load temperatures) and a highly selected absorber exit temperature (high nighttime ambient temperature), the coefficient of performance (COP) of the system would decrease. In other words, the higher the cooling load temperature and the lower the nighttime ambient temperature (and hence the lower the absorption temperature) the higher the COP of the intermittent solar absorption refrigeration cycle. Figure 4.12 shows the pure ammonia condensation/boiling (saturation) curve in a selected condenser-temperature range.

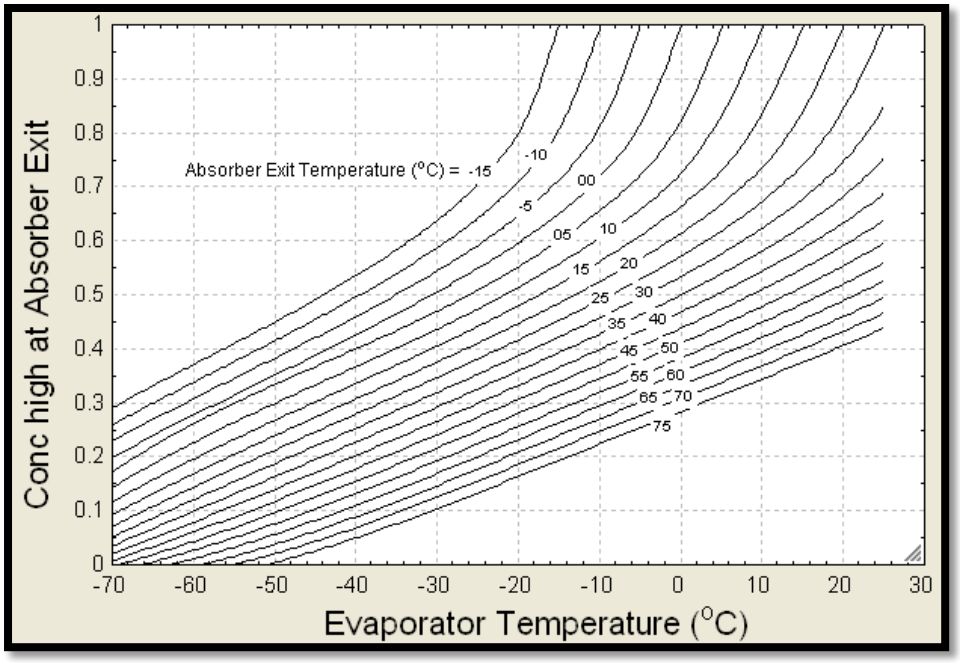

Figure 4 . 11 : Highest NH3 Concentration in aqua-ammonia (at absorber exit) v/s evaporator temperature in the absorption system

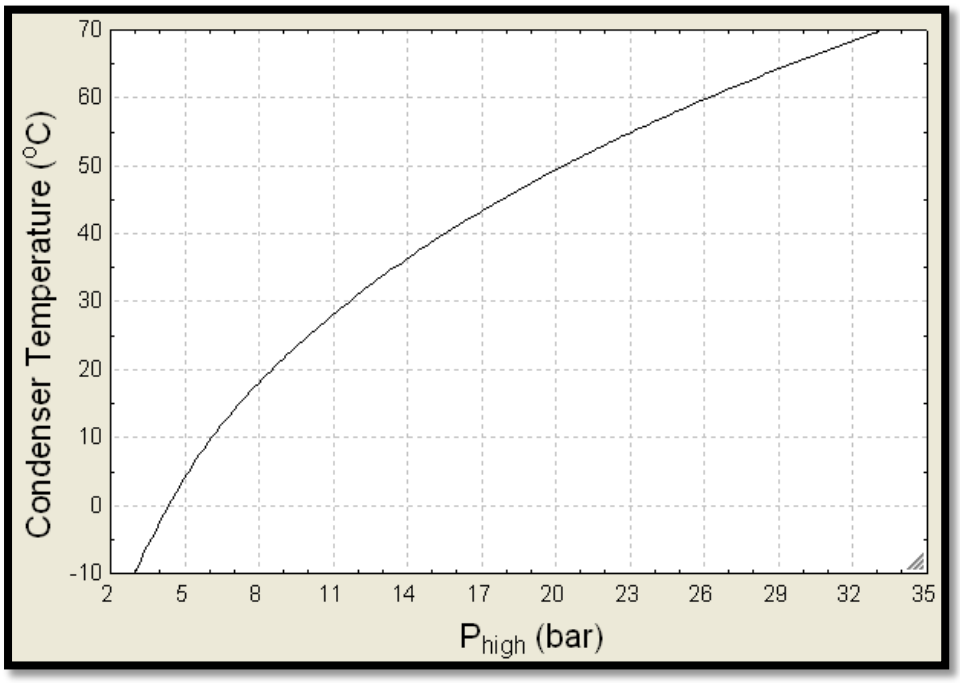

Figure 4 . 12 : Condensation Temperature v/s high pressure in the absorption system

Hence, Fig. 4.12 gives the condensation temperature against the high (condenser/generator or desorption) pressure (P_{high}) in the system. From Fig. 4.12, it is clear that for all possible condensation (daytime ambient) temperatures the condenser pressure is above atmospheric pressure (1.013 bar) for the aqua-ammonia absorption refrigeration systems. It is noteworthy, that such condenser pressures are below atmospheric pressure (vacuum pressures) in Lithium Bromide (Li Br) -water cooling/air conditioning systems, where water is the refrigerant and Li Br is the absorbent.

Figure 4.13 gives the lowest ammonia mass concentration at the generator exit against the high (condenser/generator) pressure in the system for various selected generator exit (maximum) temperatures (from 70 ˚C to 200 ˚C). It can be seen from Fig. 4.13 that, for a given generator exit (maximum) temperature (i.e. for a given solar collector type and a given solar intensity), decreasing the high (condenser/generator) pressure in the system (i.e. decreasing the daytime ambient temperature) reduces the lowest ammonia mass concentration of the aqua-ammonia solution in the absorption system.

It can also be seen that, for a given high (condenser/generator) pressure, as the generator exit temperature is increased, the lowest concentration in the system is similarly reduced. This reduction in the lowest ammonia concentration in the aqua-ammonia absorption system actually enhances the performance as it increases the concentration difference (between the highest and the lowest concentration values in the system). Thus it directly increases the ammonia generation (desorption) in the generator,

which in turn increases the refrigeration effect produced at the evaporator. Hence at very high system pressures (high daytime ambient temperature) and very low selected generator exit temperatures (ordinary non-concentrating flat-plat collector of a low performance), the lowest possible concentration value increases and hence the COP of the system decreases.

Combining Fig. 4.12 and Fig. 4.13, the required lowest ammonia mass concentration in the system (at the generator exit) against the condenser temperature (day-ambient temperature plus a few degrees) can be obtained for various selected generator exit temperatures (which depends on the available solar intensity, i.e. location and time of the year; from 70 ˚C to 200 ˚C) as shown in Fig. 4.14.

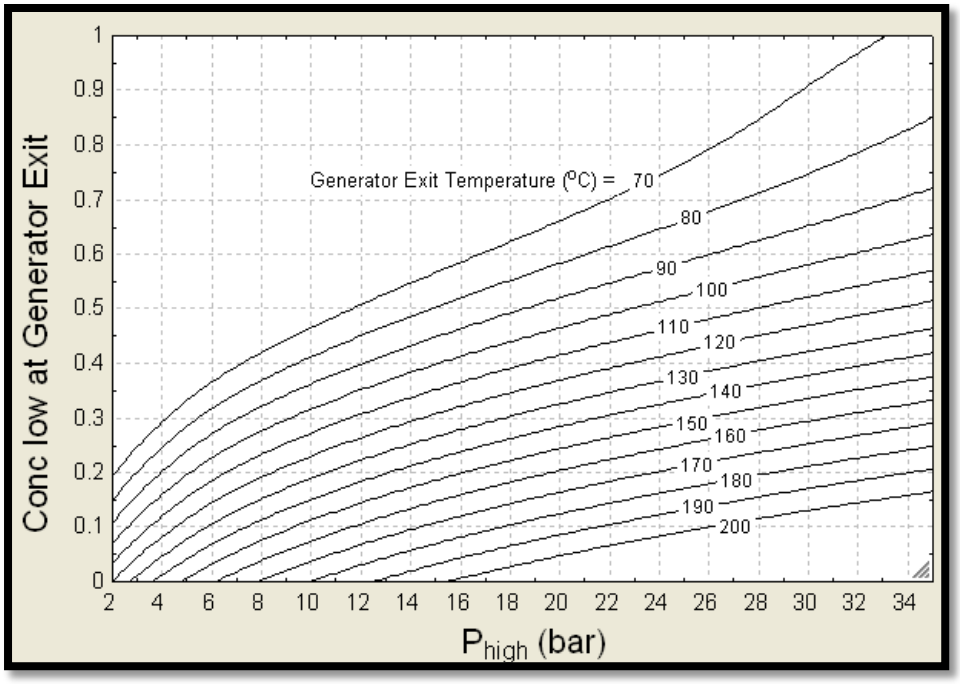

Figure 4 . 13 : Lowest ammonia concentration at the generator exit v/s high pressure in the absorption system

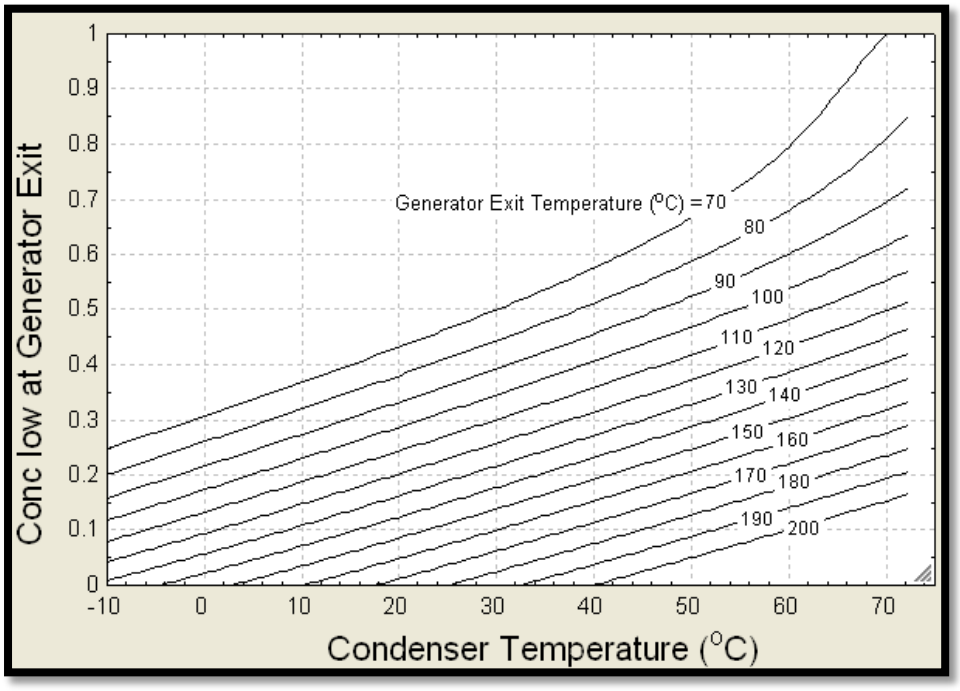

Figure 4 . 14 : Lowest Concentration at generator exit v/s condenser temperature in the absorption system

4.5 Analysis of Ambient Temperature Data for KSA

As the discussion in section 4.4 has explained in detail the importance of ambient temperatures on designing an aqua-ammonia vapor absorption refrigeration system, therefore a thorough investigation about the ambient temperatures in KSA throughout the year has been done.

Climatic Zones of Saudi Arabia

The annual variation of ambient temperature at any specific location depends upon the climate of that region. Saudi Arabia is identified with five major climatic zones [48-50]. In this study, each climatic zone is represented by one city. Climatic zone 1 is identified as subtropical with a Mediterranean subzone and a mountainous subtype. Abha (latitude = 18.23) city weather data is taken as a representative city of climatic zone 1. The climate of zone 2 is identified as hot and dry with a maritime desert subzone. Jeddah (latitude = 21.68) city weather data is taken as a representative city of climatic zone 2. The climate of zone 3 is identified as a hot and dry maritime subzone. Dhahran (latitude = 26.16) city weather data is taken as a representative city of climatic zone 3. The climate of zone 4 is identified as cold and dry with a desert subzone. Tabouk (latitude = 28.38) city weather data is taken as a representative city of climatic zone 4. The climate of zone 5 is identified as hot and dry with a desert subzone. Riyadh (latitude = 24.91) city weather data is taken as a representative city of climatic zone 5.

Hourly average weather data of 14 years (from 1970 to 1983) is used to develop the Annual Ambient Temperature graphs for the representative cities of all five climatic

zones of Saudi Arabia. The Annual Ambient Temperature graph for the Abha (representative city for the climatic zone 1) is presented in Figure 4.15. From Figure 4.15, it can be seen that the maximum summer temperature in Abha is approximately 31℃ and the minimum winter temperature in Abha is approximately 6℃. Also it can be seen that the day and night temperature difference for Abha is around 12-14 ℃ throughout the year.

Similarly, the Annual Ambient Temperature graph for the Jeddah (representative city for the climatic zone 2) is presented in Figure 4.16. From figure 4.16, it can be seen that the maximum summer temperature in Jeddah is approximately 38℃ and the minimum winter temperature in Jeddah is approximately 18℃. Also it can be seen that the day and night temperature difference for Jeddah is around 8-9 ℃ throughout the summer season and around 7-8 degree Centigrade throughout the winter season.

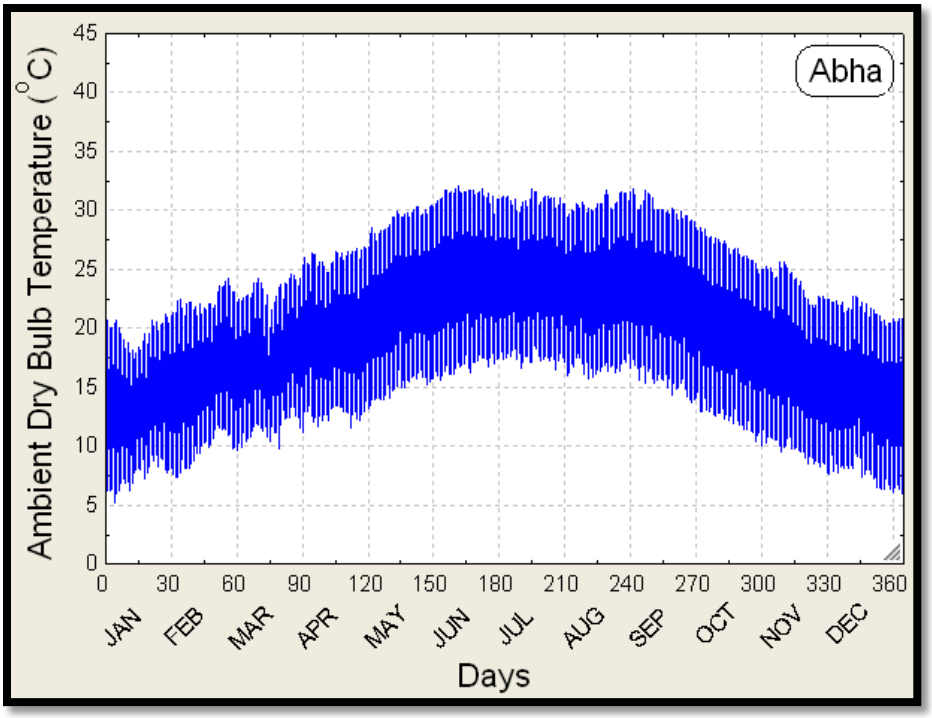

Figure 4 . 15 : Annual Ambient Temperature for ABHA

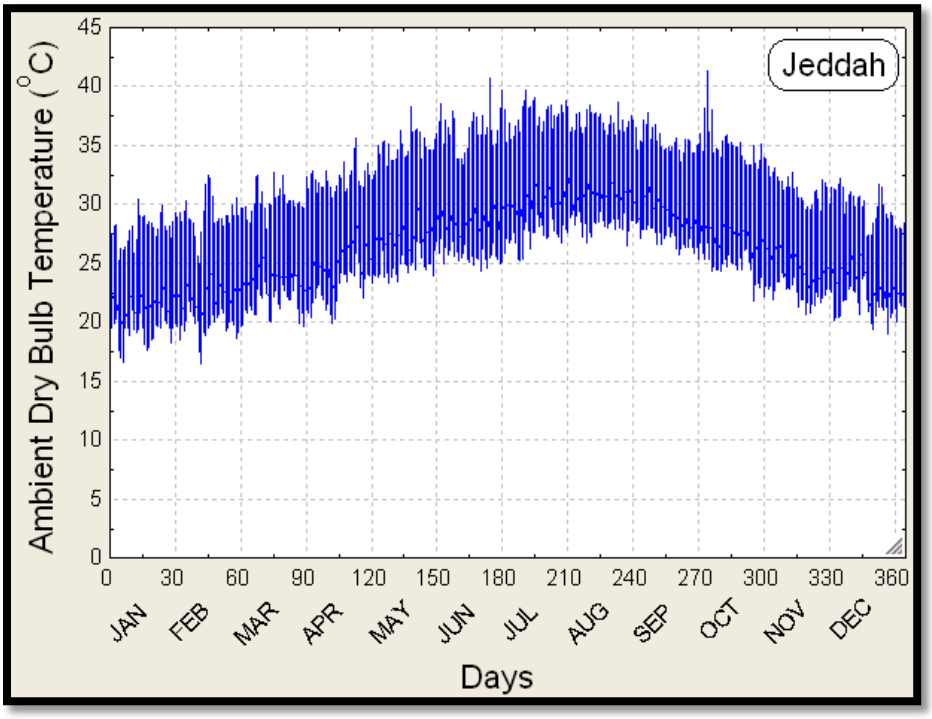

Figure 4 . 16 : Annual Ambient Temperature for JEDDAH

Similarly, the Annual Ambient Temperature graph for the Dhahran (representative city for the climatic zone 3) is presented in Figure 4.17. From Figure 4.17, it can be seen that the maximum summer temperature in Dhahran is approximately 41℃ and the minimum winter temperature in Dhahran is approximately 10℃. In addition, it can be seen that the day and night temperature difference for Dhahran is around 10-12 degree Centigrade throughout the summer season and around 7-8 degree Centigrade throughout the winter season.

Similarly, the Annual Ambient Temperature graph for the Tabouk (representative city for the climatic zone 4) is presented in Figure 4.18. From Figure 4.18, it can be seen that the maximum summer temperature in Tabouk is approximately 38℃ and the minimum winter temperature in Tabouk is approximately 03℃. Also it can be seen that the day and night temperature difference for Tabouk is around 15-17 degree Centigrade throughout the summer season and around 12-14 degree Centigrade throughout the winter season.

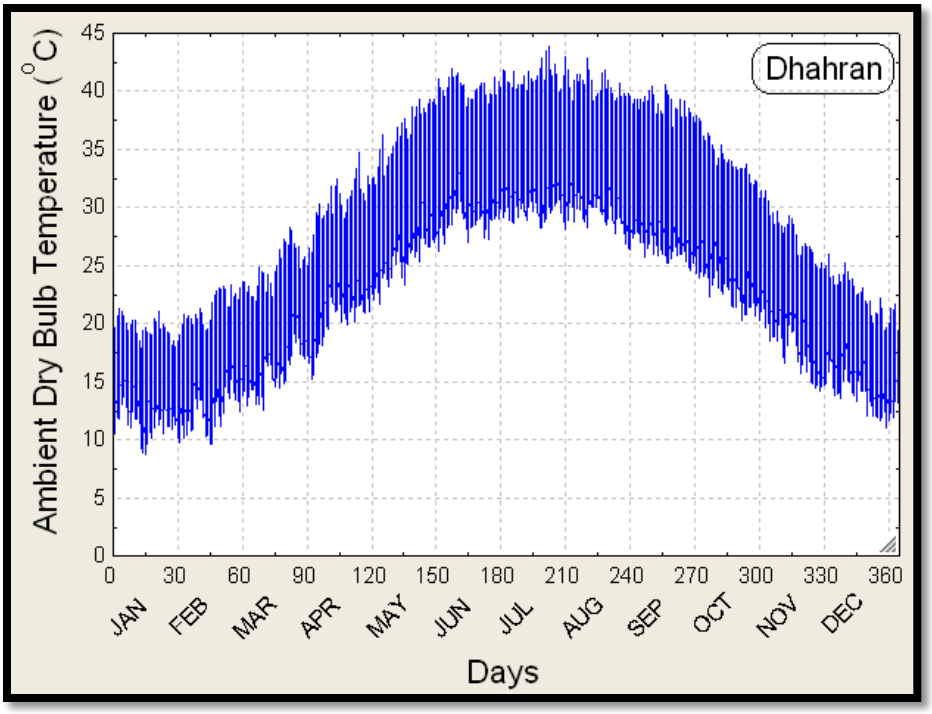

Figure 4 . 17 : Annual Ambient Temperature for DHAHRAN

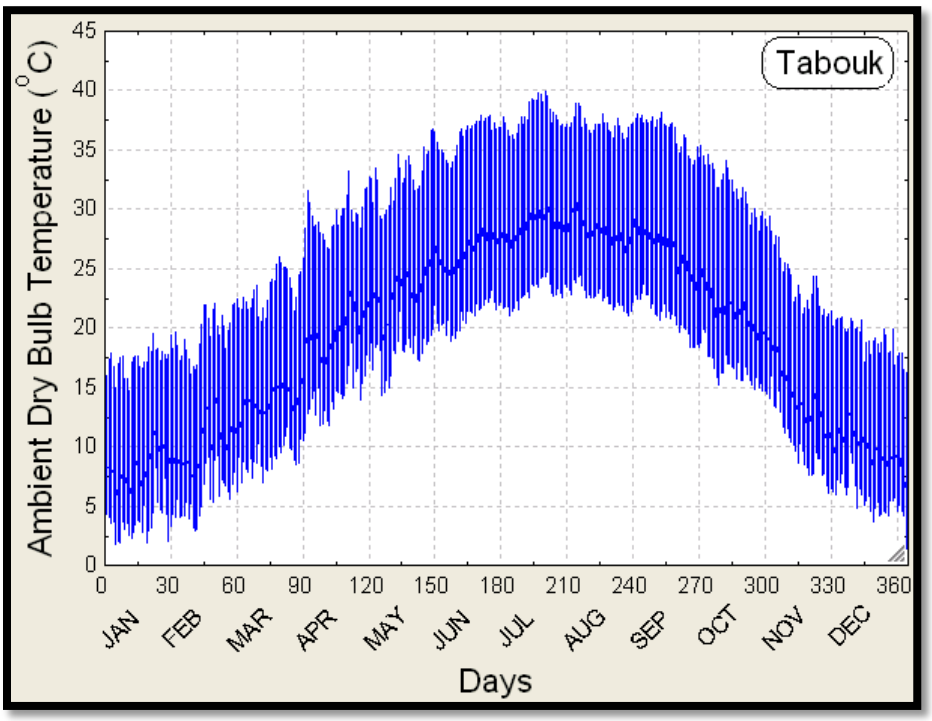

Figure 4 . 18 : Annual Ambient Temperature for TABOUK

Similarly, the Annual Ambient Temperature graph for the Riyadh (representative city for the climatic zone 5) is presented in Figure 4.19. From Figure 4.19, it can be seen that the maximum summer temperature in Riyadh is approximately 44℃ and the minimum winter temperature in Riyadh is approximately 07℃. In addition, it can be seen that the day and night temperature difference for Riyadh is around 14-16 degree Centigrade throughout the summer season and around 10-12 degree Centigrade throughout the winter season.

From the analysis of Annual Ambient Temperature graphs for the representative cities of five climatic zones of Saudi Arabia, it is visible that both Riyadh and Tabouk shows high temperature difference between summer and winter ambient temperatures, Dhahran shows medium temperature difference however Jeddah and Abha shows the least temperature difference between summer and winter seasons. Table 4.2 shows the summary of temperature data for the representative cities as an hourly average of 13 years. As the climate of Tabouk and Riyadh has high temperature difference between the day time and night time temperatures, therefore aqua-ammonia intermittent vapor absorption systems will be able to take the best advantage of such climates, however such systems are least suitable for Jeddah because of low day and night temperature differences. It can also be seen that both Riyadh and Dhahran presents the worst summer conditions in Saudi Arabia where the ambient temperatures exceeds 40℃ in peak summer season.

Hence, for our analysis, Dhahran will be considered as design station for the development of Solar Powered Aqua-Ammonia Vapor Absorption Refrigeration System.

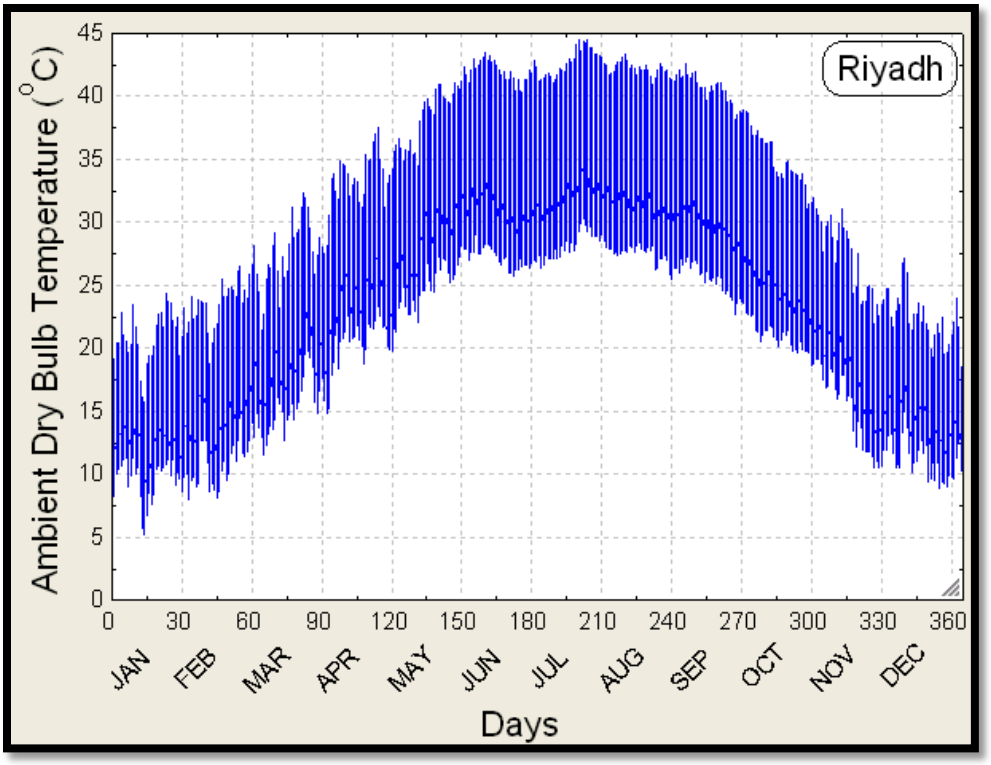

Figure 4 . 19 : Annual Ambient Temperature for RIYADH

Table 4 . 2 : Summary of Temperature Data for KSA

Climatic Zone	*Representative City*	*Maximum Summer Temperature*	*Minimum Winter Temperature*	*Summer Diurnal Temperature Difference*	*Winter Diurnal Temperature Difference*
1	*Abha*	*31 ℃*	*6 ℃*	*12-14 ℃*	*12-14 ℃*
2	*Jeddah*	*38 ℃*	*18 ℃*	*8-9 ℃*	*7-8 ℃*
3	*Dhahran*	*41 ℃*	*10 ℃*	*10-12 ℃*	*7-8 ℃*
4	*Tabouk*	*38 ℃*	*3 ℃*	*15-17 ℃*	*12-14 ℃*
5	*Riyadh*	*44 ℃*	*7 ℃*	*14-16 ℃*	*10-12 ℃*

4.6 Solar Collector Selection

As the discussion in section 4.4 has explained in detail the importance of the generator exit temperature in determining the lowest ammonia mass concentration in the system, this section explains the key in achieving that generator exit temperature. Since the generator is heated by the solar collector, so the generator exit temperature is a direct function of the mean solar collector operating temperature which; itself is a function of both, the type of solar collector selected and the thermal efficiency of selected solar collector under the ambient conditions.

The efficiency of a collector is defined as the ratio of the energy amount transferred from the collector to the heat transfer medium to the incident radiant energy on the collector. This efficiency depends on the quality of the absorber surface, the geometry of the absorber, the heat conductivity of the absorber, the transparency of the cover, and the heat losses of the collector through infrared radiation, conduction, and convection. Besides thermal loss there is always optical loss as well. The thermal loss depends upon the temperature difference between the ambient and the average temperature of the fluid in the solar collector. The higher the temperature difference, the more heat is lost. Above a specific temperature difference, the amount of heat loss equals the energy yield of the collector, so that no energy at all is delivered to the solar circulation system as shown in a typical collector efficiency curve in Figure 4.20.

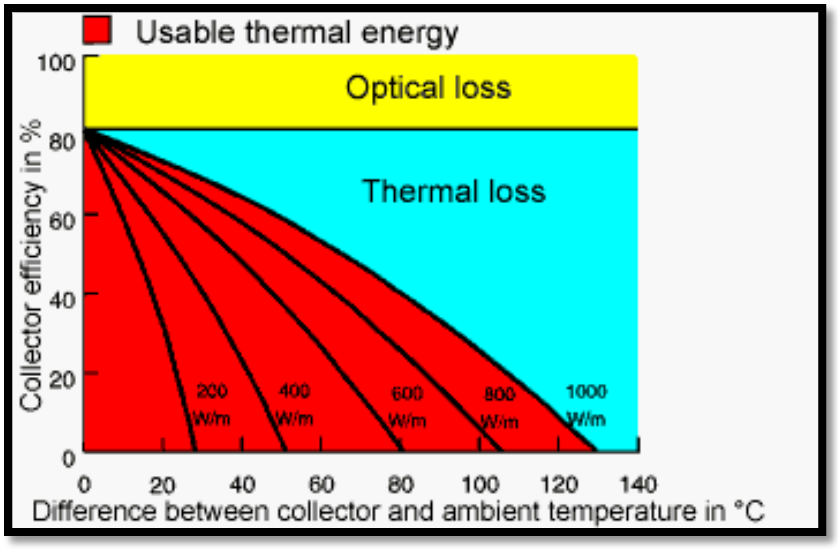

Figure 4 . 20 : Typical Collector Efficiency Curve for Flat Plate Solar Collector

Table 4 . 3 : Typical Solar Collector Characteristics

Type of Collector	Thermal Loss Factor in W/m² °C	Temperature Range in °C
Absorber (uncovered)	10 to 30	up to 40
Flat-plate collector	2,9 to 5,3	20 to 80
Evacuated-plate collector	2,6 to 4,3	20 to 120
Evacuated-tube collector	0,7 to 2,0	50 to 120
Reservoir collector	about 2,4	20 to 70
Air collector	8 to 30	20 to 50

Table 4.3 shows typical solar collector characteristics for different kind of solar collectors. Thus each type of solar collector has its own efficiency curve. A quantitative comparison indicates that the efficiency is particularly dominated by the radiation losses. The efficiency for a certain collector is not a fixed value, but is dependent on the application, e.g. temperature levels, wind speed, etc. Thus, a characteristic curve is obtained by plotting the efficiency as a function of the ratio of the temperature difference of the average temperature of the heat transfer fluid of the collector and the ambient temperature ($t_m - t_a$) to the incident radiant energy "G" as shown in Figure 4.21.

Hence, the selection of a solar collector type is a direct function of the temperature raise that is required to be achieved above ambient under the local available solar irradiance at that specific location.

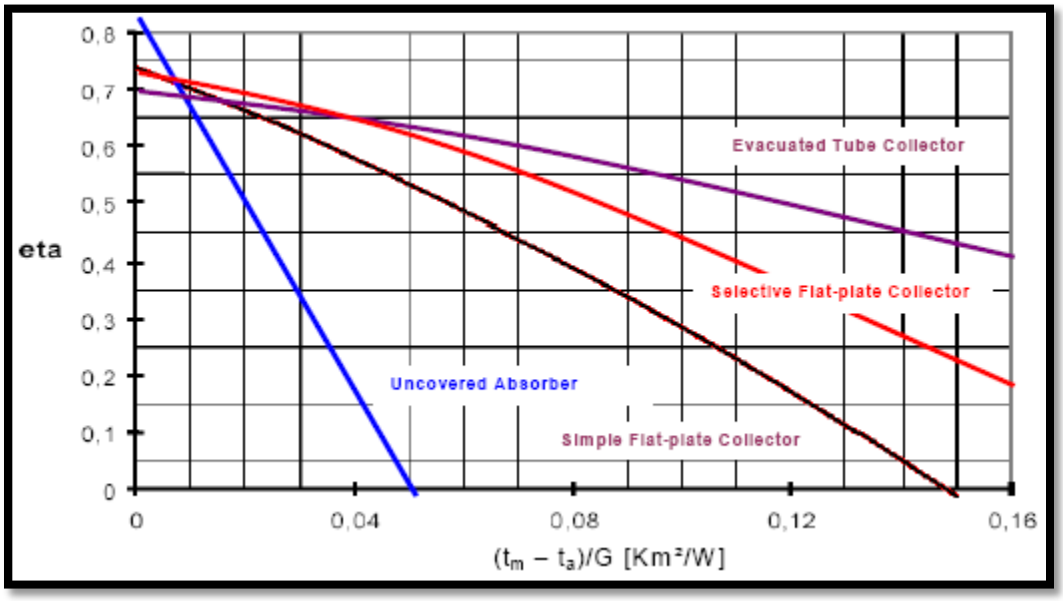

Figure 4 . 21 : Typical Collector Efficiency Curve for different kind of solar collectors

4.7 Ammonia Purification Level

As stated in section 4.3 that ammonia purification level (Highest concentration of ammonia in the system) also behaves as a basic performance parameter in determining the actual performance of the absorption system [51]. Although both the rectifier and the dephlegmator in an aqua-ammonia vapor absorption system performs the purification of ammonia before it enters into the condenser, yet it is practically not possible to 100% purify ammonia. In this regards, the setting of dephlegmator exit temperature actually determines the level of purification of ammonia from water vapors and thus determines the actual amount of ammonia going to produce the refrigeration effect at the evaporator. Dephlegmator is the unit in the vapor absorption refrigeration system that performs the complete rectification of ammonia vapors coming from the generator of the vapor absorption refrigeration system. As heat is removed from the aqua-ammonia vapors entering into the Dephlegmator from the generator, part of the aqua-ammonia vapor condenses. The condensed solution has low concentration of ammonia (i.e. having higher water content) than the aqua-ammonia vapor in the Dephlegmator. As a result rectification takes place until the concentration of ammonia in the generated vapor increases to 1 (i.e. pure ammonia) as the complete water content from the aqua-ammonia solution vapor has been removed.

Although it is desired to have pure ammonia vapor at the exit of Dephlegmator, yet from the analysis [51] it is found that it is not possible to have pure ammonia at the exit of Dephlegmator. It is because of the fact that if the Dephlegmator is designed to provide pure ammonia at the exit, the temperature at the exit of Dephlegmator must be equal to that of the temperature at the exit of condenser. In that case, knowing that the

condensation of the pure substance takes place isothermally, even a slight disturbance in the system (i.e. a little high-energy rejection at the Dephlegmator) will cause a lot of pure ammonia to be condensed inside the Dephlegmator silently.

Thus, it is not practical to design Dephlegmator exit temperature equal to the condenser exit temperature. However, the Dephlegmator exit temperature is recommended to be set close to the condensation temperature of ammonia so that in case of energy fluctuations in the Dephlegmator, the exit temperature will behave as an indicator of whether or not pure ammonia is being condensed inside the Dephlegmator. Hence the Dephlegmator exit temperature should be higher than the condenser exit temperature and as a result Dephlegmator exit concentration would always be less than 1 (i.e. pure ammonia) which means that there is always going to be a small fraction of water that is entering into the condenser.

Throughout the thermodynamic analysis in this study, the highest concentration of ammonia will be taken as 0.996. This will lead us to a temperature set at the exit of dephlegmator corresponding to the high/condenser pressure in the system thus obtaining the desired level of purification of ammonia vapors in the system as well as allowing most of ammonia vapors to enter into the condenser safely without being condensed in the dephlegmator during the ammonia purification process.

4.8 Typical Efficiency of LLHE

As stated in section 4.3, the efficiency of Liquid-Liquid Heat Exchanger (LLHE) also affects the performance of the absorption system. This heat exchanger undergoes the heat exchange between the cold strong aqua-ammonia solutions coming from the absorber exit with the hot weak aqua-ammonia solution coming from the exit of generator. Due to high value of minimum flow thermal capacitance (mCp), the heat exchange is comparatively very high. Thus, this is a heavy duty heat exchanger that undergoes the heat recovery for the system. The efficiency of this heat exchanger strongly affects the overall performance of the absorption system; therefore, it is desired to keep the efficiency of LLHE as high as possible. In this study, design of absorption system will be based on typical efficiency of LLHE equal to 0.8; however, the variation in the performance of the system with the efficiency of the LLHE will also be analyzed.

4.9 Total Cooling Load and Load Temperature

The total cooling load requirement is the basic parameter that determines the mass flow rates in the system. Since this analysis is focused on the development of a prototype that may be able to fulfill the cooling requirements of a typical family living in Dhahran, therefore, the total cooling load requirement will be taken as 5 kW for a period of 24 hours (day and night) at a refrigeration (load) temperature of -9 ℃ for freezing application. This means that the total cooling load requirement will be 120 kW-hr (i.e. enough to produce 1295 kg of ice).

4.10 Ammonia Expansion Process and Effect of VLHE

As stated in section 4.3, the ammonia expansion process and efficiency of VLHE also behaves as a basic performance parameter in determining the actual performance of the absorption system [52]. The ammonia produces the Refrigeration Effect required at the evaporator after expansion for all absorption refrigeration systems therefore; it is mainly the ammonia expansion process, which plays a key role in producing the required refrigeration effect in an aqua-ammonia vapor absorption system. Since, the overall effect of VLHE over the performance of absorption system is fundamentally small; therefore, for the purpose of analysis in this study, the typical efficiency of VLHE will be taken as 0.5.

4.11 Thermodynamic Analysis of Alternative Designs

This section discusses the detailed thermodynamic analysis for all the alternatives explained in chapter 3.

4.11.1 Cold Storage Systems (D1)

As discussed in chapter 3, the cold storage systems can be further classified in terms of their method of operation i.e. Continuous and Intermittent systems.

4.11.1.1 Continuous based Operation (D1.1)

The Continuous based operation for aqua-ammonia vapor absorption refrigeration system using cold storage can further be classified as Single Stage Systems and Double Stage Systems.

The current analysis is based on the summer conditions of Dhahran, Saudi Arabia, where the day time ambient temperature is taken as 40 ℃ as per weather condition analysis for Dhahran discussed in section 4.5. Therefore, both the absorber and the condenser is assumed to be operating at a temperature of 45 ℃ i.e. 5 ℃ higher than the ambient for single stage systems as well as for the first stage of the double stage systems. The refrigeration load is required to be maintained at a temperature of -9 ℃ and the total cooling load is required to be 120 kW-hr i.e. an average of 5 kW for 24 hours in a day for single stage systems as well as for the second stage of the double stage systems. The generator is designed to be operating at 120 ℃ i.e. 80 ℃ higher than the ambient for single stage systems however for double stage systems, the generator is designed to be operating at 80 ℃ i.e. 40 ℃ higher than the ambient. The highest concentration of ammonia required in the system is 0.996 and the typical efficiencies of LLHX, VLHE and rectifier are taken as 0.8, 0.5 and 0.75 respectively. The system is designed under the assumption that the generator can be continuously powered by the solar energy for 10 hours in a day. An EES module, given in APPENDIX A1 for single stage systems and given in APPENDIX A2 for double stage systems, has been generated to analyze the system. The results of the analysis are obtained using the mass and energy balance over the components of the absorption system. The schematic diagrams (h-t-p-x and t-p-x) of both single stage and double stage systems are given from Figure 4.22 to Figure 4.25. The thermodynamic details of each component of the single stage and double stage absorption system as given in Table 4.4 to Table 4.16: -

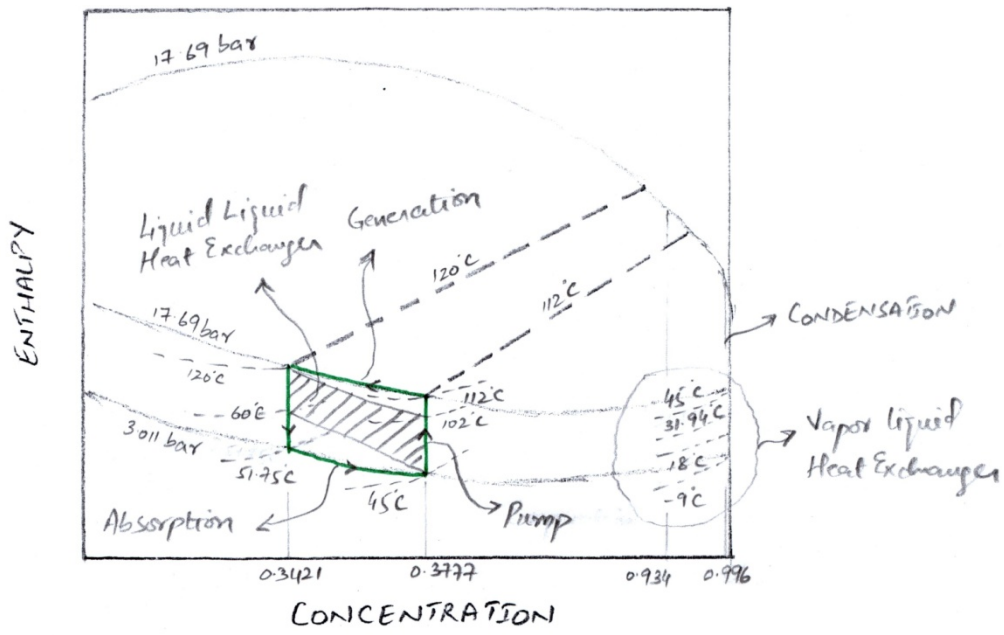

Figure 4 . 22 : Schematic h-t-p-x diagram for Alternative D1.1.1

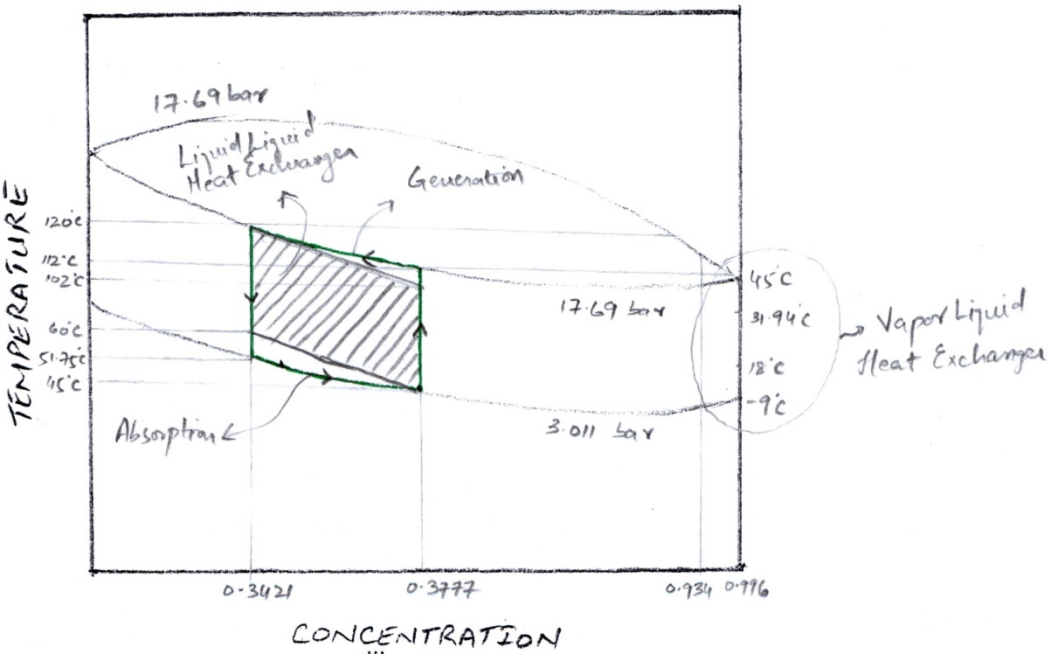

Figure 4 . 23 : Schematic t-p-x diagram for Alternative D1.1.1

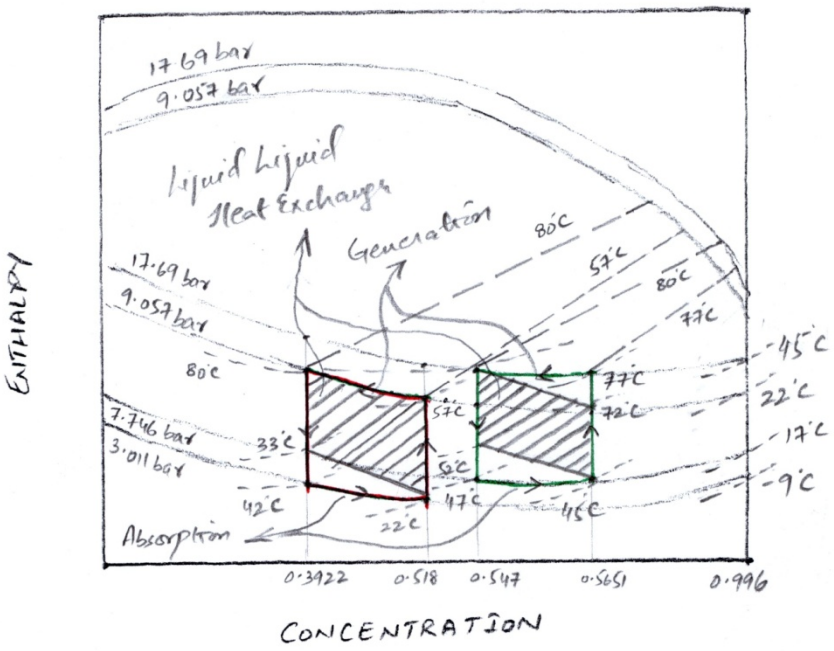

Figure 4 . 24 : Schematic h-t-p-x diagram for Alternative D1.1.2

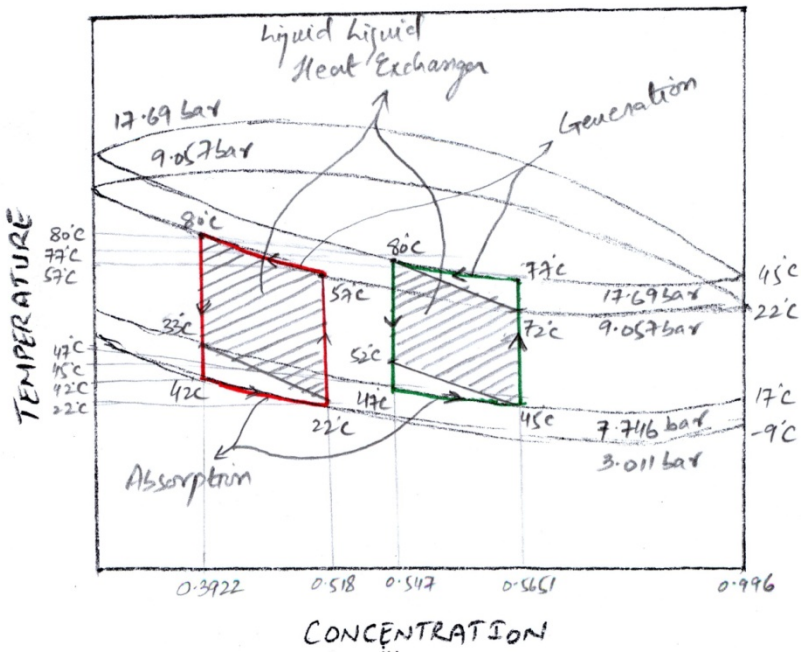

Figure 4 . 25 : Schematic t-p-x diagram for Alternative D1.1.2

Table 4 . 4 : Thermodynamic Analysis for Generator of Alternative D1.1

GENERATOR		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of Strong solution	*℃*	*102.2*	*72.01*	*57.62*
Exit Temperature of Weak solution	*℃*	*120*	*80*	*80*
Generation Starting Temperature	*℃*	*112.1*	*77.21*	*57.52*
Average Exit Temperature of Aqua-Ammonia Vapors	*℃*	*117.8*	*80.09*	*68.86*
Inlet Ammonia Mass Concentration of Strong solution	*-*	*0.3777*	*0.5651*	*0.518*
Exit Ammonia Mass Concentration of Weak solution	*-*	*0.3421*	*0.547*	*0.3922*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	*-*	*0.9336*	*0.9912*	*0.9859*
Inlet Mass Flow Rate of Strong solution	*kg/sec*	*0.1984*	*0.67*	*0.04817*
Exit Mass Flow Rate of Weak solution	*kg/sec*	*0.1876*	*0.6429*	*0.03814*
Inlet Enthalpy of Strong solution	*kJ/kg*	*234.1*	*94.29*	*21.81*
Exit Enthalpy of Weak solution	*kJ/kg*	*325.2*	*128.6*	*128.7*
Generation Starting Enthalpy	*kJ/kg*	*280.7*	*119*	*19.69*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1577*	*1413*	*1428*
Pole Generator (Enthalpy)	*kJ/kg*	*155.1*	*39.3*	*-343.8*
Exit Entropy of Weak solution	*kJ/kgK*	*1.501*	*0.9903*	*0.9949*
Generation Starting Entropy	*kJ/kgK*	*1.399*	*0.9574*	*0.6825*
Exit Specific Volume of Weak solution	*m³/kg*	*0.00127*	*0.00134*	*0.00123*
Generation Starting Specific Volume	*m³/kg*	*0.00128*	*0.00134*	*0.00127*
Heat Duty of the Generator	*kW*	*31.92*	*57.38*	*18.02*

Table 4 . 5 : Thermodynamic Analysis for Rectifier of Alternative D1.1

RECTIFIER		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Effectiveness	-	*0.75*	*0.75*	*0.75*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of the Aqua-Ammonia Vapors	*℃*	*117.8*	*80.09*	*68.86*
Exit Temperature of Aqua-Ammonia Vapors	*℃*	*106.1*	*74.03*	*60.43*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.9336*	*0.9912*	*0.9859*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.9577*	*0.9936*	*0.992*
Inlet Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1577*	*1413*	*1428*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1520*	*1393*	*1400*

Table 4 . 6 : Thermodynamic Analysis for Dephlegmator of Alternative D1.1

DEPHLEGMATOR		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of the Aqua-Ammonia Vapors	*℃*	*106.1*	*74.03*	*60.43*
Exit Temperature of Aqua-Ammonia Vapors	*℃*	*68.06*	*68.06*	*51.66*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.9577*	*0.9936*	*0.992*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.996*	*0.996*	*0.996*
Inlet Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1520*	*1393*	*1400*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1371*	*1371*	*1370*
Pole Dephlegmator (Enthalpy)	*kJ/kg*	*1605*	*1400*	*1411*
Inlet Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.765*	*4.416*	*4.731*
Exit Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.353*	*4.353*	*4.639*
Inlet Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.09542*	*0.08479*	*0.1677*
Exit Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.08241*	*0.08241*	*0.1615*
Heat Duty of the Dephlegmator	*kW*	*2.533*	*0.7832*	*0.4166*

Table 4 . 7 : Thermodynamic Analysis for Condenser of Alternative D1.1

CONDENSER		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of the Aqua-Ammonia Vapors	*℃*	*68.06*	*68.06*	*51.66*
Exit Temperature of Aqua-Ammonia Liquid	*℃*	*45*	*45*	*22*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	*-*	*0.996*	*0.996*	*0.996*
Exit Ammonia Mass Concentration of Aqua-Ammonia Liquid	*-*	*0.996*	*0.996*	*0.996*
Refrigerant Mass Flow Rate	*kg/sec*	*0.01081*	*0.02708*	*0.01004*
Inlet Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1371*	*1371*	*1370*
Exit Enthalpy of Aqua-Ammonia Liquid	*kJ/kg*	*212.3*	*212.3*	*99.55*
Inlet Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.353*	*4.353*	*4.639*
Exit Entropy of Aqua-Ammonia Liquid	*kJ/kgK*	*0.7373*	*0.7373*	*0.3742*
Inlet Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.08241*	*0.08241*	*0.1615*
Exit Specific Volume of Aqua-Ammonia Liquid	*m³/kg*	*0.00175*	*0.00175*	*0.00164*
Heat Duty of the Condenser	*kW*	*12.53*	*31.38*	*12.75*

Table 4 . 8 : Thermodynamic Analysis for VLHE of Alternative D1.1

VAPOR LIQUID HEAT EXCHANGER		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Effectiveness		*0.5*	*0.5*	*0.5*
Inlet Temperature of Saturated Liquid Ammonia	*℃*	*45*	*45*	*22*
Exit Temperature of Sub-Cooled Liquid Ammonia	*℃*	*31.94*	*37.27*	*14.06*
Inlet Temperature of Saturated Vapor Ammonia	*℃*	*-9*	*17*	*-9*
Exit Temperature of Superheated Vapor Ammonia	*℃*	*18*	*31*	*6.5*
Inlet Enthalpy of Saturated Liquid Ammonia	*kJ/kg*	*212.3*	*212.3*	*99.55*
Exit Enthalpy of Sub-Cooled Liquid Ammonia	*kJ/kg*	*147.7*	*173.8*	*61.98*
Inlet Enthalpy of Saturated Vapor Ammonia	*kJ/kg*	*1258*	*1284*	*1258*
Exit Enthalpy of Superheated Vapor Ammonia	*kJ/kg*	*1322*	*1322*	*1295*
Heat Duty of the VLHE	*kW*	*0.6984*	*1.042*	*0.377*

Table 4 . 9 : Thermodynamic Analysis for Expansion Device of Alternative D1.1

EXPANSION DEVICE		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Pressure before Expansion	*bar*	*17.69*	*17.69*	*9.057*
Pressure after Expansion	*bar*	*3.011*	*7.746*	*3.011*
Temperature before Expansion	*℃*	*31.94*	*37.27*	*14.06*
Temperature after Expansion	*℃*	*-9*	*17*	*-9*
Ammonia Mass Concentration before Expansion	*-*	*0.996*	*0.996*	*0.996*
Ammonia Mass Concentration after Expansion	*-*	*0.996*	*0.996*	*0.996*
Enthalpy before Expansion	*kJ/kg*	*147.7*	*173.8*	*61.98*
Enthalpy after Expansion	*kJ/kg*	*147.7*	*173.8*	*61.98*
Vapor Fraction after Expansion	*-*	*0.1477*	*0.08084*	*0.08175*
Entropy before Expansion	*kJ/kgK*	*0.5299*	*0.6148*	*0.245*
Entropy after Expansion	*kJ/kgK*	*0.5863*	*0.6301*	*0.2632*
Specific Volume before Expansion	*m³/kg*	*0.00169*	*0.00171*	*0.00161*
Specific Volume after Expansion	*m³/kg*	*0.06108*	*0.01481*	*0.03449*

Table 4 . 10 : Thermodynamic Analysis for Evaporator of Alternative D1.1

EVAPORATOR		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*3.011*	*7.746*	*3.011*
Operating Temperature	*℃*	*-9*	*17*	*-9*
Inlet Enthalpy of Aqua-Ammonia Liquid	*kJ/kg*	*147.7*	*173.8*	*61.98*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1258*	*1284*	*1258*
Inlet Entropy of Aqua-Ammonia Liquid	*kJ/kgK*	*0.5863*	*0.6301*	*0.2632*
Exit Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.756*	*4.427*	*4.756*
Inlet Specific Volume of Aqua-Ammonia Liquid	*m³/kg*	*0.06108*	*0.01481*	*0.03449*
Exit Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.4045*	*0.1647*	*0.4045*
Heat Duty of the Evaporator	*kW*	*12*	*30.05*	*12*

Table 4 . 11 : Thermodynamic Analysis for LLHE of Alternative D1.1

LIQUID-LIQUID HEAT EXCHANGER		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Effectiveness		*0.8*	*0.8*	*0.8*
Inlet Temperature of Saturated Weak Solution	*℃*	*120*	*80*	*80*
Exit Temperature of Sub-Cooled Weak Solution	*℃*	*60*	*52*	*33.6*
Inlet Temperature of Sub cooled Strong Solution	*℃*	*45*	*45*	*22*
Exit Temperature of Preheated Strong Solution	*℃*	*102.2*	*72.01*	*57.62*
Mass Flow Rate of Strong solution	*kg/sec*	*0.1984*	*0.67*	*0.04817*
Mass Flow Rate of Weak solution	*kg/sec*	*0.1876*	*0.6429*	*0.03814*
Inlet Enthalpy of Saturated Weak Solution	*kJ/kg*	*325.2*	*128.6*	*128.7*
Exit Enthalpy of Sub-Cooled Weak Solution	*kJ/kg*	*53.64*	*-1.897*	*-77.05*
Inlet Enthalpy of Sub cooled Strong Solution	*kJ/kg*	*-22.66*	*-30.88*	*-141.1*
Exit Enthalpy of Preheated Strong Solution	*kJ/kg*	*234.1*	*94.29*	*21.81*
Heat Duty of the LLHE	*kW*	*50.96*	*83.87*	*7.848*

Table 4 . 12 : Thermodynamic Analysis for Pump of Alternative D1.1

PUMP		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Inlet Pressure	*bar*	*3.011*	*7.746*	*3.011*
Exit Pressure	*bar*	*17.69*	*17.69*	*9.057*
Operating Temperature	*℃*	*45*	*45*	*22*
Operating Concentration	*-*	*0.3777*	*0.5651*	*0.518*
Operating Mass Flow Rate	*kg/sec*	*0.1984*	*0.67*	*0.04817*
Inlet Enthalpy	*kJ/kg*	*-23.88*	*-31.61*	*-141.7*
Exit Enthalpy	*kJ/kg*	*-22.66*	*-30.88*	*-141.1*
Inlet Entropy	*kJ/kgK*	*0.5366*	*0.5101*	*0.1684*
Exit Entropy	*kJ/kgK*	*0.5351*	*0.5048*	*0.1678*
Inlet Specific Volume	*m³/kg*	*0.00117*	*0.00128*	*0.00122*
Exit Specific Volume	*m³/kg*	*0.00117*	*0.00128*	*0.00122*
Heat Duty of the Pump (Electrical Energy Required)	*kW*	*0.2413*	*0.4836*	*0.02677*

Table 4 . 13 : Thermodynamic Analysis for Absorber of Alternative D1.1

ABSORBER		(D1.1.1.)	(D1.1.2.)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*3.011*	*7.746*	*3.011*
Inlet Temperature of Weak solution	*℃*	*60*	*52*	*33.6*
Exit Temperature of Strong solution	*℃*	*45*	*45*	*22*
Absorption Starting Temperature	*℃*	*51.75*	*47.54*	*42.36*
Inlet Temperature of Ammonia Vapors	*℃*	*18*	*31*	*6.5*
Inlet Ammonia Mass Concentration of Weak solution	*-*	*0.3421*	*0.547*	*0.3922*
Exit Ammonia Mass Concentration of Strong solution	*-*	*0.3777*	*0.5651*	*0.518*
Inlet Mass Flow Rate of Weak solution	*kg/sec*	*0.1876*	*0.6429*	*0.03814*
Exit Mass Flow Rate of Strong solution	*kg/sec*	*0.1984*	*0.67*	*0.04817*
Refrigerant Mass Flow Rate	*kg/sec*	*0.01081*	*0.02708*	*0.01004*
Inlet Enthalpy of Weak solution	*kJ/kg*	*53.64*	*-1.897*	*-77.05*
Exit Enthalpy of Strong solution	*kJ/kg*	*-23.88*	*-31.61*	*-141.7*
Absorption Starting Enthalpy	*kJ/kg*	*16.84*	*-22.93*	*-38.82*
Inlet Enthalpy of Ammonia Vapors	*kJ/kg*	*1322*	*1322*	*1295*
Exit Entropy of Strong solution	*kJ/kgK*	*0.5366*	*0.5101*	*0.1684*
Absorption Starting Entropy	*kJ/kgK*	*0.645*	*0.5442*	*0.4951*
Inlet Entropy of Ammonia Vapors	*kJ/kgK*	*4.989*	*4.555*	*4.894*
Exit Specific Volume of Strong solution	*m³/kg*	*0.00117*	*0.00128*	*0.00122*
Absorption Starting Specific Volume	*m³/kg*	*0.00116*	*0.00127*	*0.00117*
Inlet Specific Volume of Ammonia Vapors	*m³/kg*	*0.4546*	*0.1761*	*0.4336*
Heat Duty of the Absorber	*kW*	*29.1*	*55.75*	*16.88*

Table 4 . 14 : Thermodynamic Analysis for Coolant HX of Alternative D1.1

COOLANT HEAT EXCHANGER		(D1.1.1.)	(D1.1.2.)	
			First Stage	second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Heat Duty of the Coolant Heat Exchanger	*kW*	*44.16*	*87.9*	*30.05*

Table 4 . 15 : COP of Alternative D1.1

COEFFICIENT OF PERFORMANCE		(D1.1.1.)	(D1.1.2.)	
			First Stage	second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
COP		*0.372*	*0.516*	*0.6645*
			0.1573	

Table 4 . 16 : Ice Storage Requirement of Alternative D1.1

ICE STORAGE		(D1.1.1.)	(D1.1.2.)	
			First Stage	second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*No*	*Yes*
Night Time Operation		*No*	*No*	*No*
Storage Capacity Required (ICE)	*kg*	*755.4*	*0*	*755.4*

4.11.1.2 Intermittent based Operation (D1.2)

The current analysis is based on the summer conditions of Dhahran, Saudi Arabia, where the day time ambient temperature is taken as 40 ℃ and the night time ambient temperature is taken as 32 ℃ as per weather condition analysis for Dhahran discussed in section 4.5. Therefore, the absorber is assumed to be operating at a temperature of 37 ℃ i.e. 5 ℃ higher than the night time ambient and the condenser is assumed to be operating at a temperature of 45 ℃ i.e. 5 ℃ higher than the day time ambient. The refrigeration load is required to be maintained at a temperature of -9 ℃ and the total cooling load is required to be 120 kW-hr i.e. an average of 5 kW for 24 hours in a day.

The maximum temperature of generator for this alternative is designed to be at 120 ℃ i.e. 80 ℃ higher than the ambient. The highest concentration of ammonia required in the system is 0.996 and the typical efficiencies of VLHE is 0.5. The system is designed under the assumption that the generator of this system can be continuously powered by the solar energy for 10 hours in a day. An EES module, given in APPENDIX A3, has been generated to analyze the system. The results of the analysis are obtained using the mass and energy balance over the components of the absorption system. The schematic diagrams (h-t-p-x and t-p-x) of intermittent systems are given in Figure 4.26 and Figure 4.27. The thermodynamic details of each component of the cold storage based intermittent absorption system are given in Table 4.17 to Table 4.27: -

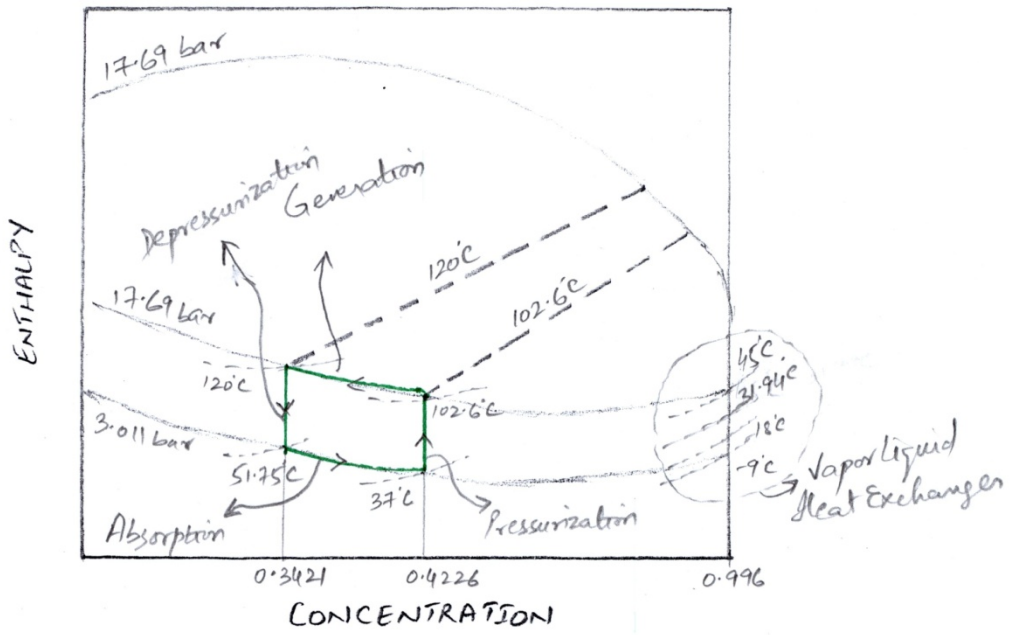

Figure 4 . 26 : Schematic h-t-p-x diagram for Alternative D1.2

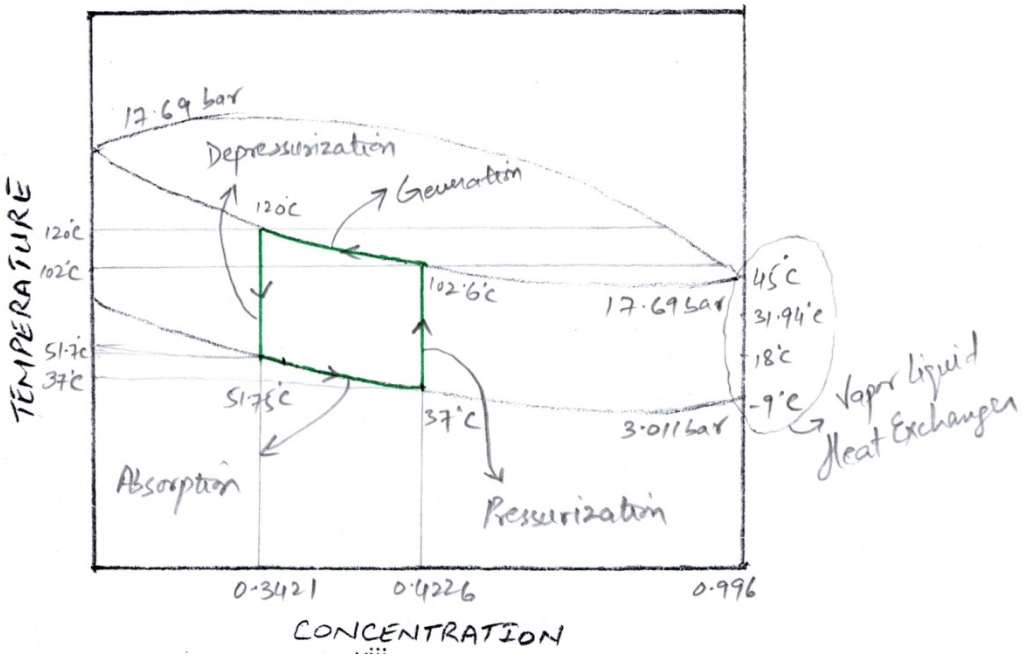

Figure 4 . 27 : Schematic t-p-x diagram for Alternative D1.2

Table 4 . 17 : Thermodynamic Analysis for Generator of Alternative D1.2

Generator (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*Yes*	
Night Time Operation	*No*	
Pressurization		
Pressure before pressurization	*3.011*	*bar*
Pressure after pressurization	*17.69*	*bar*
Pressurization Starting Temperature	*37*	*℃*
Pressurization Ending Temperature	*102.6*	*℃*
Operating Concentration	*0.4226*	*-*
Mass of Strong Solution	*3249*	*kg*
Pressurization Starting Enthalpy	*-68.94*	*kJ/kg*
Pressurization Ending Enthalpy	*230*	*kJ/kg*
Pressurization Starting Entropy	*0.4081*	*kJ/kgK*
Pressurization Ending Entropy	*1.277*	*kJ/kgK*
Pressurization Starting Specific Volume	*0.00118*	*m³/kg*
Pressurization Ending Specific Volume	*0.00129*	*m³/kg*
Energy Required for Pressurization	*971339*	*kJ*
Generation		
Operating Pressure	*17.69*	*bar*
Generation Starting Temperature	*102.6*	*℃*
Generation Ending Temperature	*120*	*℃*
Generation Starting Ammonia Mass Concentration	*0.4225*	*-*
Generation Ending Ammonia Mass Concentration	*0.3421*	*-*
Mass of Strong Solution (Generation Starting)	*3249*	*kg*
Mass of Weak Solution (Generation Ending)	*2817*	*kg*
Mass of Aqua Ammonia Vapors Generated	*431.6*	*kg*
Enthalpy of Strong Solution (Generation Starting)	*230*	*kJ/kg*
Enthalpy of Weak Solution (Generation Ending)	*325.7*	*kJ/kg*
Entropy of Weak solution (Generation Ending)	*1.501*	*kJ/kgK*
Generation Starting Entropy	*1.277*	*kJ/kgK*
Specific Volume of Weak solution (Generation Ending)	*0.00127*	*m³/kg*
Generation Starting Specific Volume	*0.00129*	*m³/kg*
Energy Required for Generation (Complete Day)	*835927*	*kJ*
Heat Duty of the Generator	*50.2*	*kW*

Table 4 . 18 : Thermodynamic Analysis for Dephlegmator of Alternative D1.2

Dephlegmator (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*Yes*	
Night Time Operation	*No*	
Operating Pressure	*17.69*	*bar*
Exit Temperature of Aqua-Ammonia Vapors	*68.06*	*℃*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.996*	*-*
Inlet Mass of Aqua Ammonia Vapors	*431.6*	*kg*
Exit Mass of Aqua Ammonia Vapors	*389.2*	*kg*
Mass of Aqua Ammonia Condensate	*42.4*	*kg*
Exit Enthalpy of Aqua-Ammonia Vapors	*1371*	*kJ/kg*
Exit Entropy of Aqua-Ammonia Vapors	*4.353*	*kJ/kgK*
Exit Specific Volume of Aqua-Ammonia Vapors	*0.08241*	*m³/kg*
Energy Required for Dephlegmation (Complete Day)	*127177*	*kJ*
Heat Duty of the Dephlegmator	*7.63831*	*kW*

Table 4 . 19 : Thermodynamic Analysis for Reflux Tank of Alternative D1.2

Reflux tank (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*Filling*	
Night Time Operation	*Emptying*	
Temperature	*95.91*	*℃*
Ammonia Mass Concentration	*0.4588*	*-*
Enthalpy	*202.4*	*kJ/kg*
Storage Capacity Required	*42.4*	*kg*

Table 4 . 20 : Thermodynamic Analysis for Condenser of Alternative D1.2

Condenser (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*Yes*	
Night Time Operation	*No*	
Operating Pressure	*17.69*	*bar*
Inlet Temperature of the Aqua-Ammonia Vapors	*68.06*	*℃*
Exit Temperature of Aqua-Ammonia Liquid	*45*	*℃*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.996*	*-*
Exit Ammonia Mass Concentration of Aqua-Ammonia Liquid	*0.996*	*-*
Condensed Mass of Aqua Ammonia Vapors	*389.2*	*kg*
Inlet Enthalpy of Aqua-Ammonia Vapors	*1371*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Liquid	*212.3*	*kJ/kg*
Inlet Entropy of Aqua-Ammonia Vapors	*4.353*	*kJ/kgK*
Exit Entropy of Aqua-Ammonia Liquid	*0.7373*	*kJ/kgK*
Inlet Specific Volume of Aqua-Ammonia Vapors	*0.08241*	*m^3/kg*
Exit Specific Volume of Aqua-Ammonia Liquid	*0.00175*	*m^3/kg*
Energy Required for Condensation (Complete Day)	*451617*	*kJ*
Heat Duty of the Condenser	*27.1114*	*kW*

Table 4 . 21 : Thermodynamic Analysis for VLHE of Alternative D1.2

Vapor Liquid Heat Exchanger (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*No*	
Night Time Operation	*Yes*	
Effectiveness	*0.5*	
Inlet Temperature of Saturated Liquid Ammonia	*45*	*℃*
Exit Temperature of Sub-Cooled Liquid Ammonia	*31.94*	*℃*
Inlet Temperature of Saturated Vapor Ammonia	*-9*	*℃*
Exit Temperature of Superheated Vapor Ammonia	*18*	*℃*
Mass of Aqua Ammonia	*389.2*	*kg*
Inlet Enthalpy of Saturated Liquid Ammonia	*212.3*	*kJ/kg*
Exit Enthalpy of Sub-Cooled Liquid Ammonia	*147.7*	*kJ/kg*
Inlet Enthalpy of Saturated Vapor Ammonia	*1258*	*kJ/kg*
Exit Enthalpy of Superheated Vapor Ammonia	*1322*	*kJ/kg*
Energy transfer (Complete Night)	*25144*	*kJ*
Heat Duty of the VLHE	*1.053*	*kW*

Table 4 . 22 : Thermodynamic Analysis for Expansion Device of Alternative D1.2

Expansion Device (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*No*	
Night Time Operation	*Yes*	
Pressure before Expansion	*17.69*	*bar*
Pressure after Expansion	*3.011*	*bar*
Temperature before Expansion	*31.94*	*℃*
Temperature after Expansion	*-9*	*℃*
Ammonia Mass Concentration before Expansion	*0.996*	*-*
Ammonia Mass Concentration after Expansion	*0.996*	*-*
Enthalpy before Expansion	*147.7*	*kJ/kg*
Enthalpy after Expansion	*147.7*	*kJ/kg*
Vapor Fraction after Expansion	*0.1477*	*-*
Entropy before Expansion	*0.5299*	*kJ/kgK*
Entropy after Expansion	*0.5863*	*kJ/kgK*
Specific Volume before Expansion	*0.00169*	*m³/kg*
Specific Volume after Expansion	*0.06108*	*m³/kg*

Table 4 . 23 : Thermodynamic Analysis for Evaporator of Alternative D1.2

Evaporator (D1.2)		
parameter	*Value*	*unit*
Day Time Operation	*No*	
Night Time Operation	*Yes*	
Operating Pressure	*3.011*	*bar*
Operating Temperature	*-9*	*℃*
Inlet Enthalpy of Aqua-Ammonia Liquid	*147.7*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Vapors	*1258*	*kJ/kg*
Inlet Entropy of Aqua-Ammonia Liquid	*0.5863*	*kJ/kgK*
Exit Entropy of Aqua-Ammonia Vapors	*4.756*	*kJ/kgK*
Inlet Specific Volume of Aqua-Ammonia Liquid	*0.06108*	*m³/kg*
Exit Specific Volume of Aqua-Ammonia Vapors	*0.4045*	*m³/kg*
Heat Duty of the Evaporator	*18.1*	*kW*

Table 4 . 24 : Thermodynamic Analysis for Absorber of Alternative D1.2

Absorber (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*No*	
Night Time Operation	*Yes*	
De-Pressurization		
Pressure before de-pressurization	*17.69*	*bar*
Pressure after de-pressurization	*3.011*	*bar*
De-pressurization Starting Temperature	*120*	*℃*
De-pressurization Ending Temperature	*51.75*	*℃*
Operating Concentration	*0.3421*	*-*
Mass of Weak Solution	*2817*	*kg*
De-pressurization Starting Enthalpy	*325.7*	*kJ/kg*
De-pressurization Ending Enthalpy	*16.84*	*kJ/kg*
De-pressurization Starting Entropy	*1.501*	*kJ/kgK*
De-pressurization Ending Entropy	*0.645*	*kJ/kgK*
De-pressurization Starting Specific Volume	*0.00127*	*m³/kg*
De-pressurization Ending Specific Volume	*0.00116*	*m³/kg*
Energy Rejected for De-pressurization	*868805*	*kJ*
Absorption		
Operating Pressure	*3.011*	*bar*
Absorption Starting Temperature	*51.75*	*℃*
Absorption Ending Temperature	*37*	*℃*
Absorption Starting Ammonia Mass Concentration	*0.3421*	*-*
Absorption Ending Ammonia Mass Concentration	*0.4225*	*-*
Mass of Weak Solution (Absorption Starting)	*2817*	*kg*
Mass of Strong Solution (Absorption Ending)	*3206.2*	*kg*
Mass of Aqua Ammonia Vapors Absorbed	*389.2*	*kg*
Enthalpy of Weak Solution (Absorption Starting)	*16.84*	*kJ/kg*
Enthalpy of Strong Solution (Absorption Ending)	*-68.94*	*kJ/kg*
Entropy of Strong solution (Absorption Ending)	*0.4081*	*kJ/kgK*
Absorption Starting Entropy	*0.645*	*kJ/kgK*
Specific Volume of Strong solution (Absorption Ending)	*0.00118*	*m³/kg*
Absorption Starting Specific Volume	*0.00116*	*m³/kg*
Energy Required for Absorption (Complete Night)	*781780*	*kJ*
Heat Duty of the Absorber	*32.75*	*kW*

Table 4 . 25 : Thermodynamic Analysis for Coolant HX of Alternative D1.2

Coolant Heat Exchanger (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*Yes*	
Night Time Operation	*Yes*	
Heat Duty of the Coolant Heat Exchanger (Day time)	*34.75*	*kW*
Heat Duty of the Coolant Heat Exchanger (Night time)	*32.75*	*kW*

Table 4 . 26 : Coefficient of Performance of Alternative D1.2

Coefficient of Performance (D1.2.)		
parameter	*Value*	*unit*
Day Time Operation	*Yes*	
Night Time Operation	*Yes*	
COP	*0.23*	

Table 4 . 27 : Ice Storage Requirement of Alternative D1.2

ICE STORAGE (D1.2.1.)		
parameter	*Value*	*unit*
Day Time Operation	*No*	
Night Time Operation	*Yes*	
Storage Capacity Required (ICE)	*539.6*	*kg*

4.11.2 Refrigerant Storage Systems (D2)

The Continuous based operation for aqua-ammonia vapor absorption refrigeration system using refrigerant storage can further be classified as Single Stage Systems and Double Stage Systems. The current analysis is based on the summer conditions of Dhahran, Saudi Arabia, where the day time ambient temperature is taken as 40 ℃ and the night time ambient temperature is taken as 32 ℃ as per weather condition analysis for Dhahran discussed in section 4.5. Therefore, at the night time, absorber/condenser is assumed to be operating at a temperature of 37 ℃ i.e. 5 ℃ higher than the night time ambient and at the day time, the absorber/condenser is assumed to be operating at a temperature of 45 ℃ i.e. 5 ℃ higher than the day time ambient, for single stage systems as well as for the first stage of the double stage systems.

The refrigeration load is required to be maintained at a temperature of -9 ℃ and the total cooling load is required to be 120 kW-hr i.e. an average of 5 kW for 24 hours in a day for single stage systems as well as for the second stage of the double stage systems. The generator is designed to be operating at 120 ℃ i.e. 80 ℃ higher than the ambient for single stage systems however for double stage systems, the generator is designed to be operating at 80 ℃ i.e. 40 ℃ higher than the ambient. The highest concentration of ammonia required in the system is 0.996 and the typical efficiencies of LLHE, VLHE and rectifier are 0.8, 0.5 and 0.75 respectively. The system is designed under the assumption

that the generator of this system can be continuously powered by the solar energy for 10 hours in a day.

An EES module, given in APPENDIX A4 for single stage systems and given in APPENDIX A5 for double stage systems, has been generated to analyze the system. The results of the analysis are obtained using the mass and energy balance over the components of the absorption system. The schematic diagrams (h-t-p-x and t-p-x) of both single stage and double stage systems are given from Figure 4.28 to Figure 4.31. The thermodynamic details of each component of the single stage and double stage absorption system based on refrigerant storage system is given in Table 4.28 to Table 4.42:-

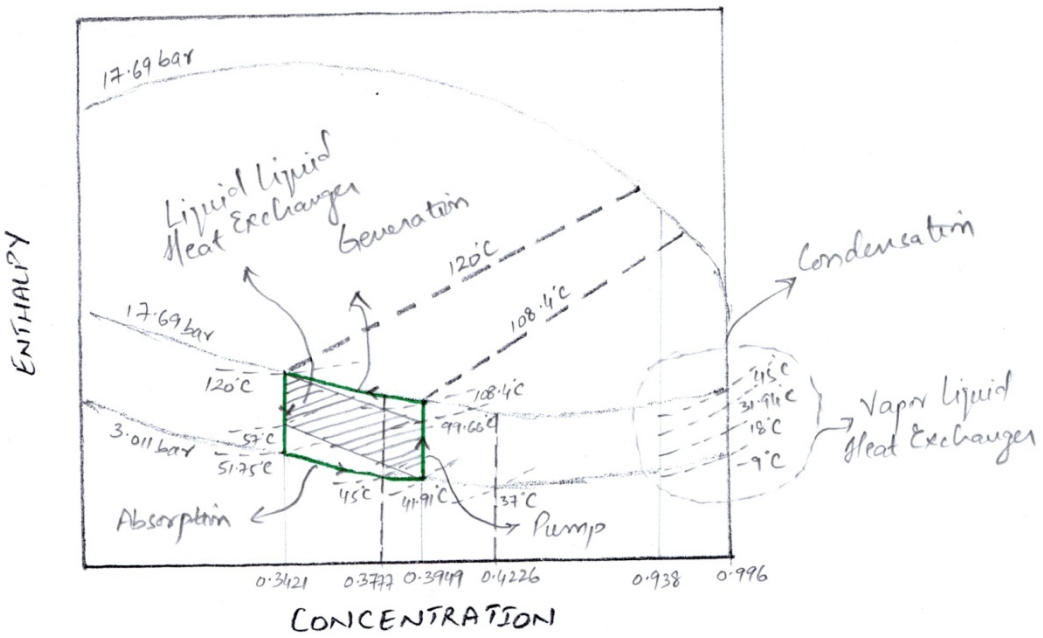

Figure 4 . 28 : Schematic h-t-p-x diagram for Alternative D2.1

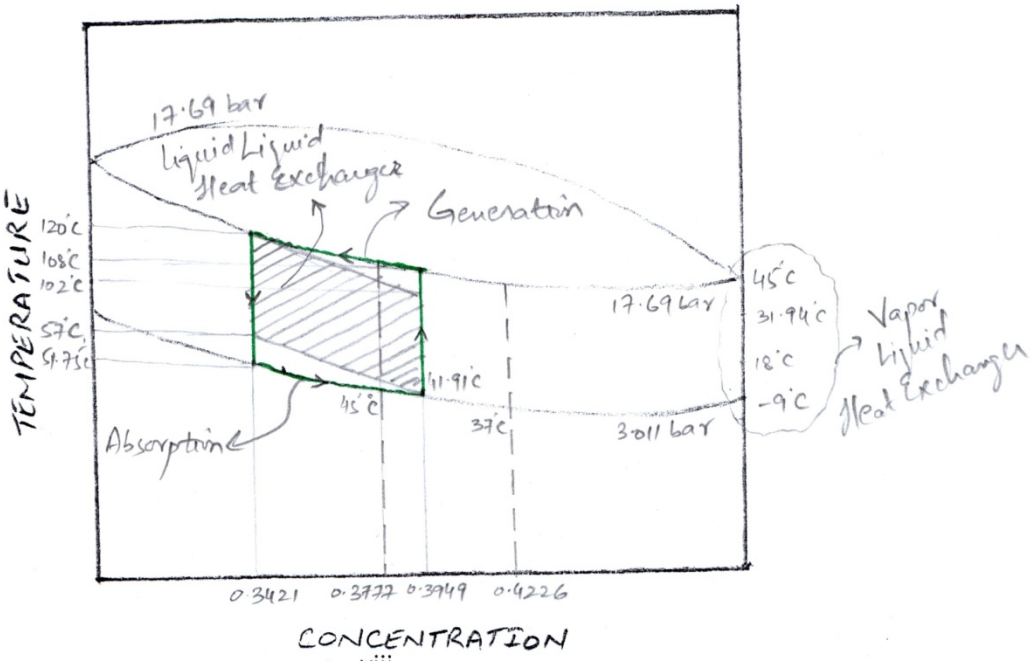

Figure 4 . 29 : Schematic t-p-x diagram for Alternative D2.1

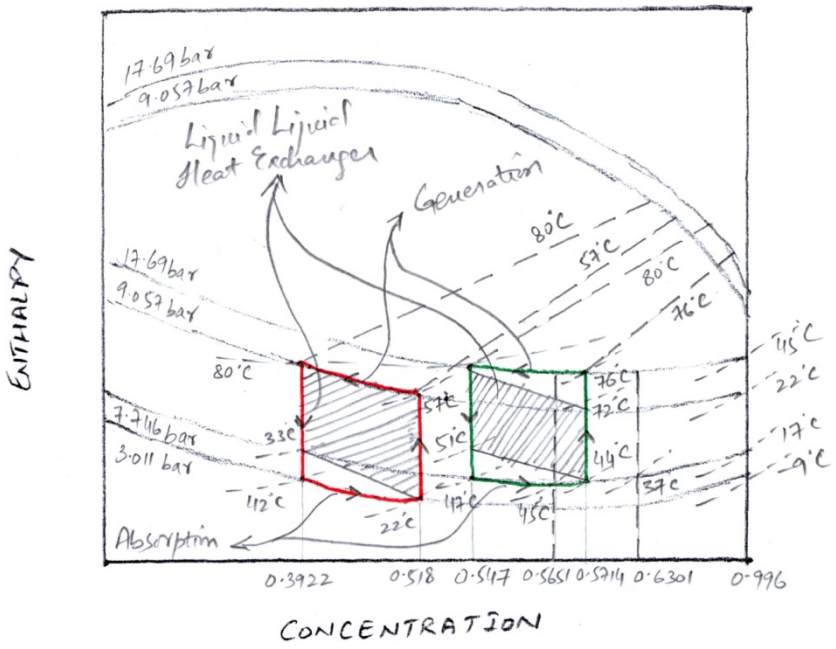

Figure 4 . 30 : Schematic h-t-p-x diagram for Alternative D2.2

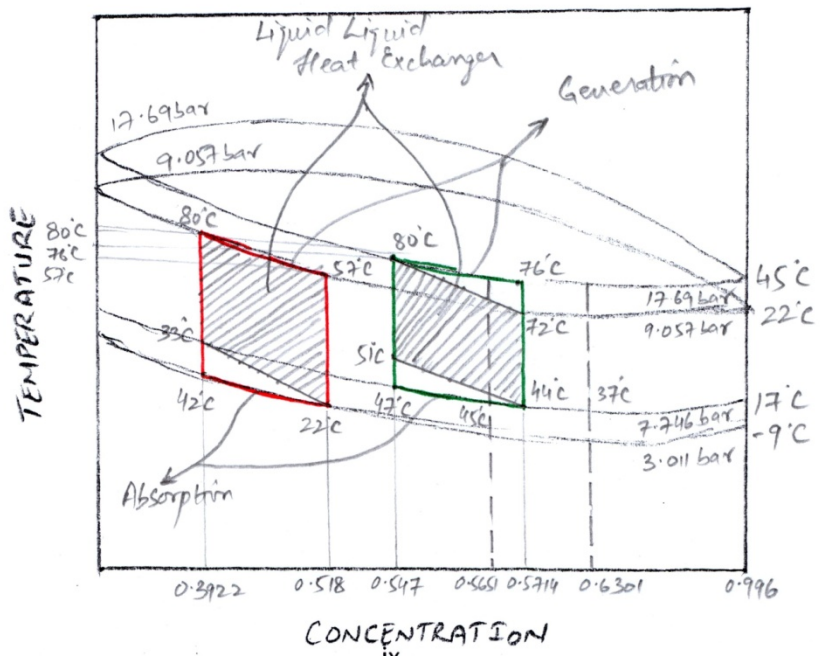

Figure 4 . 31 : Schematic t-p-x diagram for Alternative D2.2

Table 4 . 28 : Thermodynamic Analysis for Generator of Alternative D2

GENERATOR		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of Strong solution	*℃*	*99.66*	*72.01*	*57.62*
Exit Temperature of Weak solution	*℃*	*120*	*80*	*80*
Generation Starting Temperature	*℃*	*108.4*	*76.28*	*57.52*
Exit Temperature of Aqua-Ammonia Vapors	*℃*	*115.8*	*79.56*	*68.86*
Inlet Ammonia Mass Concentration of Strong solution	*-*	*0.3949*	*0.5714*	*0.518*
Exit Ammonia Mass Concentration of Weak solution	*-*	*0.3421*	*0.547*	*0.3922*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	*-*	*0.9388*	*0.9915*	*0.9859*
Inlet Mass Flow Rate of Strong solution	*kg/sec*	*0.1339*	*0.4983*	*0.04817*
Exit Mass Flow Rate of Weak solution	*kg/sec*	*0.1231*	*0.4713*	*0.03814*
Inlet Enthalpy of Strong solution	*kJ/kg*	*219.1*	*92.77*	*21.81*
Exit Enthalpy of Weak solution	*kJ/kg*	*325.2*	*128.6*	*128.7*
Generation Starting Enthalpy	*kJ/kg*	*260.5*	*116*	*19.69*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1566*	*1411*	*1428*
Pole Generator (Enthalpy)	*kJ/kg*	*99.13*	*17.82*	*-343.8*
Exit Entropy of Weak solution	*kJ/kgK*	*1.501*	*0.9903*	*0.9949*
Generation Starting Entropy	*kJ/kgK*	*1.351*	*0.9464*	*0.6825*
Exit Specific Volume of Weak solution	*m³/kg*	*0.00127*	*0.00134*	*0.00123*
Generation Starting Specific Volume	*m³/kg*	*0.00129*	*0.00135*	*0.00127*
Heat Duty of the Generator	*kW*	*27.83*	*52.19*	*18.02*

Table 4 . 29 : Thermodynamic Analysis for Rectifier of Alternative D2

RECTIFIER		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Effectiveness	-	*0.75*	*0.75*	*0.75*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of the Aqua-Ammonia Vapors	*℃*	*115.8*	*79.56*	*68.86*
Exit Temperature of Aqua-Ammonia Vapors	*℃*	*103.7*	*73.43*	*60.43*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.9388*	*0.9915*	*0.9859*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.9624*	*0.9939*	*0.992*
Inlet Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1566*	*1411*	*1428*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1509*	*1391*	*1400*

Table 4 . 30 : Thermodynamic Analysis for Dephlegmator of Alternative D2

DEPHLEGMATOR		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of the Aqua-Ammonia Vapors	*℃*	*103.7*	*73.43*	*60.43*
Exit Temperature of Aqua-Ammonia Vapors	*℃*	*68.06*	*68.06*	*51.66*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.9624*	*0.9939*	*0.992*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	-	*0.996*	*0.996*	*0.996*
Inlet Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1509*	*1391*	*1400*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1371*	*1371*	*1370*
Pole Dephlegmator (Enthalpy)	*kJ/kg*	*1585*	*1397*	*1411*
Inlet Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.736*	*4.411*	*4.731*
Exit Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.353*	*4.353*	*4.639*
Inlet Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.09469*	*0.08457*	*0.1677*
Exit Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.08241*	*0.08241*	*0.1615*
Heat Duty of the Dephlegmator	*kW*	*2.314*	*0.7073*	*0.4166*

Table 4 . 31 : Thermodynamic Analysis for Condenser of Alternative D2

CONDENSER		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Operating Pressure	*bar*	*17.69*	*17.69*	*9.057*
Inlet Temperature of the Aqua-Ammonia Vapors	*℃*	*68.06*	*68.06*	*51.66*
Exit Temperature of Aqua-Ammonia Liquid	*℃*	*45*	*45*	*22*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	*-*	*0.996*	*0.996*	*0.996*
Exit Ammonia Mass Concentration of Aqua-Ammonia Liquid	*-*	*0.996*	*0.996*	*0.996*
Refrigerant Mass Flow Rate	*kg/sec*	*0.01081*	*0.02708*	*0.01004*
Inlet Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1371*	*1371*	*1370*
Exit Enthalpy of Aqua-Ammonia Liquid	*kJ/kg*	*212.3*	*212.3*	*99.55*
Inlet Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.353*	*4.353*	*4.639*
Exit Entropy of Aqua-Ammonia Liquid	*kJ/kgK*	*0.7373*	*0.7373*	*0.3742*
Inlet Specific Volume of Aqua-Ammonia Vapors	*m³/kg*	*0.08241*	*0.08241*	*0.1615*
Exit Specific Volume of Aqua-Ammonia Liquid	*m³/kg*	*0.00175*	*0.00175*	*0.00164*
Heat Duty of the Condenser	*kW*	*12.53*	*31.38*	*12.75*

Table 4 . 32 : Refrigerant Storage Requirement of Alternative D2

REFRIGERANT STORAGE		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Filling*	*Filling*	*Filling*
Night Time Operation		*Emptying*	*Emptying*	*Emptying*
Storage Pressure	*bar*	*17.69*	*17.69*	*9.057*
Storage Temperature	*℃*	*45*	*45*	*22*
Storage Concentration	*-*	*0.996*	*0.996*	*0.996*
Storage Enthalpy	*kJ/kg*	*212.3*	*212.3*	*99.55*
Mass Flow Rate (Daytime Filling)	*kg/sec*	*0.00631*	*0.00888*	*0.00586*
Mass Flow Rate (Night time Emptying)	*kg/sec*	*0.0045*	*0.00634*	*0.00418*
Storage Capacity Required (Liquid Ammonia)	*kg*	*227*	*319.5*	*210.7*

Table 4 . 33 : Thermodynamic Analysis for VLHE of Alternative D2

VAPOR LIQUID HEAT EXCHANGER		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*Yes*	*Yes*	*Yes*
Effectiveness		*0.5*	*0.5*	*0.5*
Inlet Temperature of Saturated Liquid Ammonia	*℃*	*45*	*45*	*22*
Exit Temperature of Sub-Cooled Liquid Ammonia	*℃*	*31.94*	*37.27*	*14.06*
Inlet Temperature of Saturated Vapor Ammonia	*℃*	*-9*	*17*	*-9*
Exit Temperature of Superheated Vapor Ammonia	*℃*	*18*	*31*	*6.5*
Refrigerant Mass Flow Rate (Day Time)	*kg/sec*	*0.0045*	*0.0182*	*0.00418*
Refrigerant Mass Flow Rate (Night Time)	*kg/sec*	*0.0045*	*0.00634*	*0.00418*
Inlet Enthalpy of Saturated Liquid Ammonia	*kJ/kg*	*212.3*	*212.3*	*99.55*
Exit Enthalpy of Sub-Cooled Liquid Ammonia	*kJ/kg*	*147.7*	*173.8*	*61.98*
Inlet Enthalpy of Saturated Vapor Ammonia	*kJ/kg*	*1258*	*1284*	*1258*
Exit Enthalpy of Superheated Vapor Ammonia	*kJ/kg*	*1322*	*1322*	*1295*
Heat Duty of the VLHE (Day Time)	*kW*	*0.291*	*0.7003*	*0.1571*
Heat Duty of the VLHE (Night Time)	*kW*	*0.291*	*0.2439*	*0.1571*

Table 4 . 34 : Thermodynamic Analysis for Evaporator of Alternative D2

EVAPORATOR		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*Yes*	*Yes*	*Yes*
Operating Pressure	*bar*	*3.011*	*7.746*	*3.011*
Operating Temperature	*℃*	*-9*	*17*	*-9*
Refrigerant Mass Flow Rate (Day Time)	*kg/sec*	*0.0045*	*0.0182*	*0.00418*
Refrigerant Mass Flow Rate (Night Time)	*kg/sec*	*0.0045*	*0.00634*	*0.00418*
Inlet Enthalpy of Aqua-Ammonia Liquid	*kJ/kg*	*147.7*	*173.8*	*61.98*
Exit Enthalpy of Aqua-Ammonia Vapors	*kJ/kg*	*1258*	*1284*	*1258*
Inlet Entropy of Aqua-Ammonia Liquid	*kJ/kgK*	*0.5863*	*0.6301*	*0.2632*
Exit Entropy of Aqua-Ammonia Vapors	*kJ/kgK*	*4.756*	*4.427*	*4.756*
Inlet Specific Volume of Aqua-Ammonia Liquid	*m^3/kg*	*0.06108*	*0.01481*	*0.03449*
Exit Specific Volume of Aqua-Ammonia Vapors	*m^3/kg*	*0.4045*	*0.1647*	*0.4045*
Heat Duty of the Evaporator (Day Time)	*kW*	*5*	*20.2*	*5*
Heat Duty of the Evaporator (Night Time)	*kW*	*5*	*7.035*	*5*

Table 4 . 35 : Thermodynamic Analysis for Expansion Device of Alternative D2

EXPANSION DEVICE		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*Yes*	*Yes*	*Yes*
Pressure before Expansion	*bar*	*17.69*	*17.69*	*9.057*
Pressure after Expansion	*bar*	*3.011*	*7.746*	*3.011*
Temperature before Expansion	*℃*	*31.94*	*37.27*	*14.06*
Temperature after Expansion	*℃*	*-9*	*17*	*-9*
Ammonia Mass Concentration before Expansion	*-*	*0.996*	*0.996*	*0.996*
Ammonia Mass Concentration after Expansion	*-*	*0.996*	*0.996*	*0.996*
Refrigerant Mass Flow Rate (Day Time)	*kg/sec*	*0.0045*	*0.0182*	*0.00418*
Refrigerant Mass Flow Rate (Night Time)	*kg/sec*	*0.0045*	*0.00634*	*0.00418*
Enthalpy before Expansion	*kJ/kg*	*147.7*	*173.8*	*61.98*
Enthalpy after Expansion	*kJ/kg*	*147.7*	*173.8*	*61.98*
Vapor Fraction after Expansion	*-*	*0.1477*	*0.08084*	*0.08175*
Entropy before Expansion	*kJ/kgK*	*0.5299*	*0.6148*	*0.245*
Entropy after Expansion	*kJ/kgK*	*0.5863*	*0.6301*	*0.2632*
Specific Volume before Expansion	*m³/kg*	*0.00169*	*0.00171*	*0.00161*
Specific Volume after Expansion	*m³/kg*	*0.06108*	*0.01481*	*0.03449*

Table 4 . 36 : Weak Solution Storage Requirement of Alternative D2

WEAK SOLUTION STORAGE		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Filling*	*Filling*	*Filling*
Night Time Operation		*Emptying*	*Emptying*	*Emptying*
Storage Pressure	*bar*	*17.69*	*17.69*	*17.69*
Storage Temperature	*℃*	*57.53*	*51.35*	*33.6*
Storage Concentration	*-*	*0.3421*	*0.547*	*0.3922*
Storage Enthalpy	*kJ/kg*	*42.85*	*-4.883*	*-77.05*
Mass Flow Rate (Daytime Filling)	*kg/sec*	*0.04494*	*0.03908*	*0.02225*
Mass Flow Rate (Night time Emptying)	*kg/sec*	*0.0321*	*0.02792*	*0.01589*
Storage Capacity Required (Weak Solution)	*kg*	*1618*	*1407*	*800.9*

Table 4 . 37 : Thermodynamic Analysis for LLHE of Alternative D2

LIQUID-LIQUID HEAT EXCHANGER		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Effectiveness		*0.8*	*0.8*	*0.8*
Inlet Temperature of Saturated Weak Solution	*℃*	*120*	*80*	*80*
Exit Temperature of Sub-Cooled Weak Solution	*℃*	*57.53*	*51.35*	*33.6*
Inlet Temperature of Sub cooled Strong Solution	*℃*	*41.91*	*44.18*	*22*
Exit Temperature of Preheated Strong Solution	*℃*	*99.66*	*71.39*	*57.62*
Mass Flow Rate of Strong solution	*kg/sec*	*0.1339*	*0.4983*	*0.04817*
Mass Flow Rate of Weak solution	*kg/sec*	*0.1231*	*0.4713*	*0.03814*
Inlet Enthalpy of Saturated Weak Solution	*kJ/kg*	*325.2*	*128.6*	*128.7*
Exit Enthalpy of Sub-Cooled Weak Solution	*kJ/kg*	*42.85*	*-4.883*	*-77.05*
Inlet Enthalpy of Sub cooled Strong Solution	*kJ/kg*	*-40.48*	*-33.54*	*-141.7*
Exit Enthalpy of Preheated Strong Solution	*kJ/kg*	*219.1*	*92.77*	*21.81*
Heat Duty of the LLHE	*kW*	*34.76*	*62.88*	*7.848*

Table 4 . 38 : Strong Solution Storage Requirement of Alternative D2

STRONG SOLUTION STORAGE		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Emptying*	*Emptying*	*Emptying*
Night Time Operation		*Filling*	*Filling*	*Filling*
Storage Pressure	*bar*	*3.011*	*7.746*	*3.011*
Storage Temperature	*℃*	*37*	*37*	*22*
Storage Concentration	*-*	*0.4226*	*0.6301*	*0.518*
Storage Enthalpy	*kJ/kg*	*-68.94*	*-51.67*	*-141.7*
Mass Flow Rate (Daytime Emptying)	*kg/sec*	*0.05124*	*0.04796*	*0.0281*
Mass Flow Rate (Night time Filling)	*kg/sec*	*0.0366*	*0.03426*	*0.02007*
Storage Capacity Required (Strong Solution)	*kg*	*1845*	*1726*	*1012*

Table 4 . 39 : Thermodynamic Analysis for Absorber of Alternative D2

ABSORBER		(D2.1)	(D2.2)	
			First Stage	second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*Yes*	*Yes*	*Yes*
Operating Pressure	*bar*	*3.011*	*7.746*	*3.011*
Inlet Temperature of Weak solution	*℃*	*57.53*	*51.35*	*33.6*
Exit Temperature of Strong solution (Day time)	*℃*	*45*	*45*	*22*
Exit Temperature of Strong solution (Night time)	*℃*	*37*	*37*	*22*
Absorption Starting Temperature	*℃*	*51.75*	*47.54*	*42.36*
Inlet Temperature of Ammonia Vapors	*℃*	*18*	*31*	*6.5*
Inlet Ammonia Mass Concentration of Weak solution	*-*	*0.3421*	*0.547*	*0.3922*
Exit Ammonia Mass Concentration of Strong solution (Day time)	*-*	*0.3777*	*0.5651*	*0.518*
Exit Ammonia Mass Concentration of Strong solution (Night time)	*-*	*0.4226*	*0.6301*	*0.518*
Inlet Mass Flow Rate of Weak solution (Day time)	*kg/sec*	*0.07818*	*0.4322*	*0.01589*
Inlet Mass Flow Rate of Weak solution (Night time)	*kg/sec*	*0.0321*	*0.0279*	*0.01589*
Exit Mass Flow Rate of Strong solution (Day time)	*kg/sec*	*0.08268*	*0.4504*	*0.02007*
Exit Mass Flow Rate of Strong solution (Night time)	*kg/sec*	*0.0366*	*0.0342*	*0.02007*
Refrigerant Mass Flow Rate (Day time)	*kg/sec*	*0.04504*	*0.0182*	*0.00418*
Refrigerant Mass Flow Rate (Night time)	*kg/sec*	*0.04504*	*0.0063*	*0.00418*
Inlet Enthalpy of Weak solution	*kJ/kg*	*42.85*	*-4.883*	*-77.05*
Exit Enthalpy of Strong solution (Day time)	*kJ/kg*	*-23.88*	*-31.61*	*-141.7*
Exit Enthalpy of Strong solution (Night time)	*kJ/kg*	*-68.94*	*-51.67*	*-141.7*
Absorption Starting Enthalpy	*kJ/kg*	*16.84*	*-22.93*	*-38.82*
Inlet Enthalpy of Ammonia Vapors	*kJ/kg*	*1322*	*1322*	*1295*
Exit Entropy of Strong solution (Day time)	*kJ/kgK*	*0.5366*	*0.5101*	*0.1684*
Exit Entropy of Strong solution (Night time)	*kJ/kgK*	*0.4081*	*0.4099*	*0.1684*
Absorption Starting Entropy	*kJ/kgK*	*0.645*	*0.5442*	*0.4951*
Inlet Entropy of Ammonia Vapors	*kJ/kgK*	*4.989*	*4.555*	*4.894*
Exit Specific Volume of Strong solution (Day time)	*m³/kg*	*0.00117*	*0.0012*	*0.00122*
Exit Specific Volume of Strong solution (Night time)	*m³/kg*	*0.00118*	*0.0013*	*0.00122*
Absorption Starting Specific Volume	*m³/kg*	*0.00116*	*0.0012*	*0.00117*
Inlet Specific Volume of Ammonia Vapors	*m³/kg*	*0.4546*	*0.1761*	*0.4336*
Heat Duty of the Absorber (Day time)	*kW*	*11.28*	*36.19*	*7.035*
Heat Duty of the Absorber (Night time)	*kW*	*9.855*	*10.02*	*7.035*

Table 4 . 40 : Thermodynamic Analysis for Pump of Alternative D2

PUMP		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*No*	*No*	*No*
Inlet Pressure	*bar*	*3.011*	*7.746*	*3.011*
Exit Pressure	*bar*	*17.69*	*17.69*	*9.057*
Operating Temperature	*℃*	*41.91*	*44.18*	*22*
Operating Concentration	*-*	*0.3949*	*0.5714*	*0.518*
Operating Mass Flow Rate	*kg/sec*	*0.1339*	*0.4983*	*0.04817*
Inlet Enthalpy	*kJ/kg*	*-41.12*	*-33.54*	*-141.7*
Exit Enthalpy	*kJ/kg*	*-40.48*	*-33.42*	*-141.1*
Inlet Entropy	*kJ/kgK*	*0.4889*	*0.5013*	*0.1684*
Exit Entropy	*kJ/kgK*	*0.4855*	*0.4977*	*0.1678*
Inlet Specific Volume	*m³/kg*	*0.00136*	*0.00137*	*0.00122*
Exit Specific Volume	*m³/kg*	*0.00117*	*0.00128*	*0.00122*
Heat Duty of the Pump (Electrical Energy Required)	*kW*	*0.08557*	*0.05866*	*0.02677*

Table 4 . 41 : Thermodynamic Analysis for Coolant HX of Alternative D2

COOLANT HEAT EXCHANGER		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*Yes*	*Yes*	*Yes*
Heat Duty of the Coolant Heat Exchanger (Day time)	*kW*	*26.124*	*68.2773*	*20.2016*
Heat Duty of the Coolant Heat Exchanger (Night time)	*kW*	*9.855*	*10.02*	*7.035*

Table 4 . 42 : Coefficient of Performance of Alternative D2

COEFFICIENT OF PERFORMANCE		(D2.1)	(D2.2)	
			First Stage	Second Stage
PARAMETER	UNIT	Value	Value	Value
Day Time Operation		*Yes*	*Yes*	*Yes*
Night Time Operation		*Yes*	*No*	*No*
COP (Overall)		*0.427*	*0.5684*	*0.6645*
			0.1692	

4.11.3 Heat Storage Systems (D3)

The current analysis is based on the summer conditions of Dhahran, Saudi Arabia, where the day time ambient temperature is taken as 40 ℃ and the night time ambient temperature is taken as 32 ℃ as per weather condition analysis for Dhahran discussed in section 4.5. Therefore, at the day time, both the absorber and the condenser is assumed to be operating at a temperature of 45 ℃ i.e. 5 ℃ higher than the day time ambient temperature and at the night time, both the absorber and the condenser is assumed to be operating at a temperature of 37 ℃ i.e. 5 ℃ higher than the night time ambient temperature.

The refrigeration load is required to be maintained at a temperature of -9 ℃ and the total cooling load is required to be 120 kW-hr i.e. an average of 5 kW for 24 hours in a day. The generator of this alternative is designed to be operating at 80 ℃ higher than the ambient. The highest concentration of ammonia required in the system is 0.996 and the typical efficiencies of LLHX, VLHE and rectifier are 0.8, 0.5 and 0.75 respectively. The system is designed under the assumption that the generator of this system can be continuously powered by the solar energy for 10 hours in a day. An EES module, given in APPENDIX A6, has been generated to analyze the system. Generator is used to produce the aqua ammonia vapors for refrigeration. The schematic diagrams (h-t-p-x and t-p-x) are given in Figure 4.32 and 4.33. The thermodynamic details of heat storage based absorption refrigeration system are given in Table 4.43 to Table 4.55.

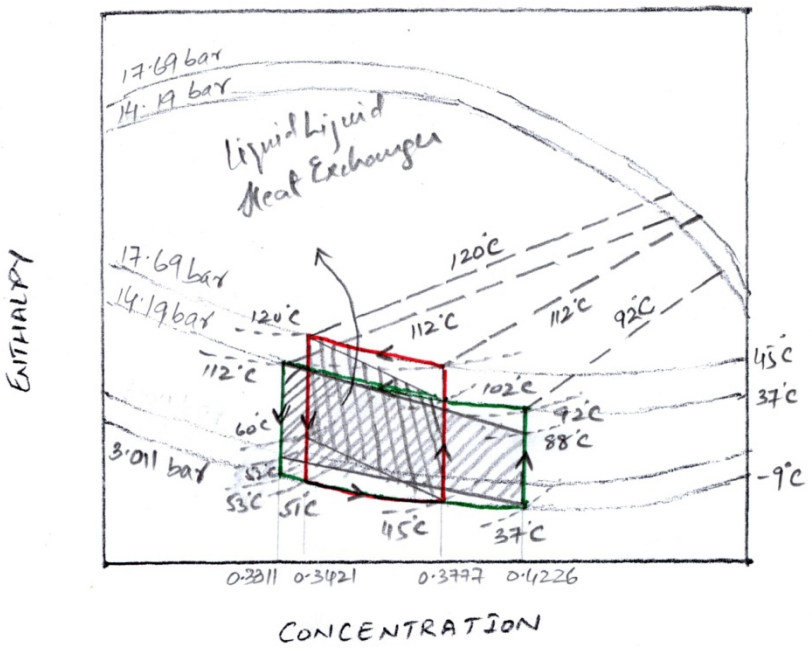

Figure 4 . 32 : Schematic h-t-p-x diagram for Alternative D3

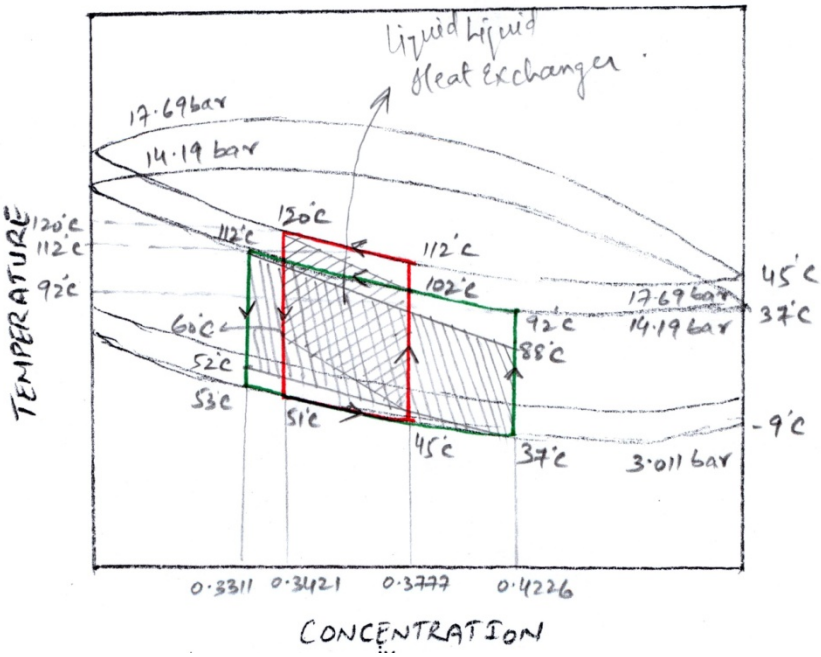

Figure 4 . 33 : Schematic t-p-x diagram for Alternative D3

Table 4 . 43 : Thermodynamic Analysis for Generator of Alternative D3

Generator (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Operating Pressure	*17.69*	*bar*	*14.19*	*bar*
Inlet Temperature of Strong solution	*102.2*	*℃*	*88.45*	*℃*
Exit Temperature of Weak solution	*120*	*℃*	*112*	*℃*
Generation Starting Temperature	*112.1*	*℃*	*92.57*	*℃*
Exit Temperature of Aqua-Ammonia Vapors	*117.8*	*℃*	*103.3*	*℃*
Inlet Ammonia Mass Concentration of Strong solution	*0.3777*	*-*	*0.4226*	*-*
Exit Ammonia Mass Concentration of Weak solution	*0.3421*	*-*	*0.3311*	*-*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.9336*	*-*	*0.952*	*-*
Inlet Mass Flow Rate of Strong solution	*0.08268*	*kg/sec*	*0.03189*	*kg/sec*
Exit Mass Flow Rate of Weak solution	*0.07818*	*kg/sec*	*0.0275*	*kg/sec*
Inlet Enthalpy of Strong solution	*234.1*	*kJ/kg*	*163.2*	*kJ/kg*
Exit Enthalpy of Weak solution	*325.2*	*kJ/kg*	*290.4*	*kJ/kg*
Generation Starting Enthalpy	*280.7*	*kJ/kg*	*182.6*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Vapors	*1577*	*kJ/kg*	*1532*	*kJ/kg*
Pole Generator (Enthalpy)	*155.1*	*kJ/kg*	*-56.54*	*kJ/kg*
Exit Entropy of Weak solution	*1.501*	*kJ/kgK*	*1.408*	*kJ/kgK*
Generation Starting Entropy	*1.399*	*kJ/kgK*	*1.15*	*kJ/kgK*
Exit Specific Volume of Weak solution	*0.00127*	*m³/kg*	*0.00125*	*m³/kg*
Generation Starting Specific Volume	*0.00128*	*m³/kg*	*0.00127*	*m³/kg*
Heat Duty of the Generator	*13.3*	*kW*	*9.54*	*kW*

Table 4 . 44 : Thermodynamic Analysis for Rectifier of Alternative D3

Rectifier (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Effectiveness	*0.75*	*-*	*0.75*	*-*
Operating Pressure	*17.69*	*bar*	*14.19*	*bar*
Inlet Temperature of the Aqua-Ammonia Vapors	*117.8*	*℃*	*103.3*	*℃*
Exit Temperature of Aqua-Ammonia Vapors	*106.1*	*℃*	*92.17*	*℃*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.9336*	*-*	*0.952*	*-*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.9577*	*-*	*0.9712*	*-*
Inlet Enthalpy of Aqua-Ammonia Vapors	*1577*	*kJ/kg*	*1532*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Vapors	*1520*	*kJ/kg*	*1482*	*kJ/kg*

Table 4 . 45 : Thermodynamic Analysis for Dephlegmator of Alternative D3

Dephlegmator (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Operating Pressure	*17.69*	*bar*	*14.19*	*bar*
Inlet Temperature of the Aqua-Ammonia Vapors	*106.1*	*℃*	*92.17*	*℃*
Exit Temperature of Aqua-Ammonia Vapors	*68.06*	*℃*	*62.3*	*℃*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.9577*	*-*	*0.9712*	*-*
Exit Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.996*	*-*	*0.996*	*-*
Inlet Enthalpy of Aqua-Ammonia Vapors	*1520*	*kJ/kg*	*1482*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Vapors	*1371*	*kJ/kg*	*1367*	*kJ/kg*
Pole Dephlegmator (Enthalpy)	*1605*	*kJ/kg*	*1542*	*kJ/kg*
Inlet Entropy of Aqua-Ammonia Vapors	*4.765*	*kJ/kgK*	*4.763*	*kJ/kgK*
Exit Entropy of Aqua-Ammonia Vapors	*4.353*	*kJ/kgK*	*4.433*	*kJ/kgK*
Inlet Specific Volume of Aqua-Ammonia Vapors	*0.09542*	*m³/kg*	*0.1158*	*m³/kg*
Exit Specific Volume of Aqua-Ammonia Vapors	*0.08241*	*m³/kg*	*0.1024*	*m³/kg*
Heat Duty of the Dephlegmator	*1.055*	*kW*	*0.7668*	*kW*

Table 4 . 46 : Thermodynamic Analysis for Condenser of Alternative D3

Condenser (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Operating Pressure	*17.69*	*bar*	*14.19*	*bar*
Inlet Temperature of the Aqua-Ammonia Vapors	*68.06*	*℃*	*62.3*	*℃*
Exit Temperature of Aqua-Ammonia Liquid	*45*	*℃*	*37*	*℃*
Inlet Ammonia Mass Concentration of Aqua-Ammonia Vapors	*0.996*	*-*	*0.996*	*-*
Exit Ammonia Mass Concentration of Aqua-Ammonia Liquid	*0.996*	*-*	*0.996*	*-*
Refrigerant Mass Flow Rate	*0.0045*	*kg/sec*	*0.00438*	*kg/sec*
Inlet Enthalpy of Aqua-Ammonia Vapors	*1371*	*kJ/kg*	*1367*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Liquid	*212.3*	*kJ/kg*	*172.5*	*kJ/kg*
Inlet Entropy of Aqua-Ammonia Vapors	*4.353*	*kJ/kgK*	*4.433*	*kJ/kgK*
Exit Entropy of Aqua-Ammonia Liquid	*0.7373*	*kJ/kgK*	*0.6122*	*kJ/kgK*
Inlet Specific Volume of Aqua-Ammonia Vapors	*0.08241*	*m³/kg*	*0.1024*	*m³/kg*
Exit Specific Volume of Aqua-Ammonia Liquid	*0.00175*	*m³/kg*	*0.00171*	*m³/kg*
Heat Duty of the Condenser	*5.22*	*kW*	*5.236*	*kW*

Table 4 . 47 : Thermodynamic Analysis for VLHE of Alternative D3

Vapor Liquid Heat Exchanger (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Effectiveness	*0.5*		*0.5*	
Inlet Temperature of Saturated Liquid Ammonia	*45*	*℃*	*37*	*℃*
Exit Temperature of Sub-Cooled Liquid Ammonia	*31.94*	*℃*	*25.64*	*℃*
Inlet Temperature of Saturated Vapor Ammonia	*-9*	*℃*	*-9*	*℃*
Exit Temperature of Superheated Vapor Ammonia	*18*	*℃*	*14*	*℃*
Inlet Enthalpy of Saturated Liquid Ammonia	*212.3*	*kJ/kg*	*172.5*	*kJ/kg*
Exit Enthalpy of Sub-Cooled Liquid Ammonia	*147.7*	*kJ/kg*	*117.2*	*kJ/kg*
Inlet Enthalpy of Saturated Vapor Ammonia	*1258*	*kJ/kg*	*1258*	*kJ/kg*
Exit Enthalpy of Superheated Vapor Ammonia	*1322*	*kJ/kg*	*1313*	*kJ/kg*
Heat Duty of the VLHE	*0.291*	*kW*	*0.2422*	*kW*

Table 4 . 48 : Thermodynamic Analysis for Expansion Device of Alternative D3

Expansion Device (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Pressure before Expansion	*17.69*	*bar*	*14.19*	*bar*
Pressure after Expansion	*3.011*	*bar*	*3.011*	*bar*
Temperature before Expansion	*31.94*	*℃*	*25.64*	*℃*
Temperature after Expansion	*-9*	*℃*	*-9*	*℃*
Ammonia Mass Concentration before Expansion	*0.996*	*-*	*0.996*	*-*
Ammonia Mass Concentration after Expansion	*0.996*	*-*	*0.996*	*-*
Enthalpy before Expansion	*147.7*	*kJ/kg*	*117.2*	*kJ/kg*
Enthalpy after Expansion	*147.7*	*kJ/kg*	*117.2*	*kJ/kg*
Vapor Fraction after Expansion	*0.1477*	*-*	*0.1242*	*-*
Entropy before Expansion	*0.5299*	*kJ/kgK*	*0.4307*	*kJ/kgK*
Entropy after Expansion	*0.5863*	*kJ/kgK*	*0.4712*	*kJ/kgK*
Specific Volume before Expansion	*0.00169*	*m³/kg*	*0.00166*	*m³/kg*
Specific Volume after Expansion	*0.06108*	*m³/kg*	*0.05161*	*m³/kg*

Table 4 . 49 : Thermodynamic Analysis for Evaporator of Alternative D3

Evaporator (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Operating Pressure	*3.011*	*bar*	*3.011*	*bar*
Operating Temperature	*-9*	*℃*	*-9*	*℃*
Inlet Enthalpy of Aqua-Ammonia Liquid	*147.7*	*kJ/kg*	*117.2*	*kJ/kg*
Exit Enthalpy of Aqua-Ammonia Vapors	*1258*	*kJ/kg*	*1258*	*kJ/kg*
Inlet Entropy of Aqua-Ammonia Liquid	*0.5863*	*kJ/kgK*	*0.4712*	*kJ/kgK*
Exit Entropy of Aqua-Ammonia Vapors	*4.756*	*kJ/kgK*	*4.756*	*kJ/kgK*
Inlet Specific Volume of Aqua-Ammonia Liquid	*0.06108*	*m³/kg*	*0.05161*	*m³/kg*
Exit Specific Volume of Aqua-Ammonia Vapors	*0.4045*	*m³/kg*	*0.4045*	*m³/kg*
Heat Duty of the Evaporator	*5*	*kW*	*5*	*kW*

Table 4 . 50 : Thermodynamic Analysis for LLHE of Alternative D3

Liquid-Liquid Heat Exchanger (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Effectiveness	*0.8*		*0.8*	
Inlet Temperature of Saturated Weak Solution	*120*	*℃*	*112*	*℃*
Exit Temperature of Sub-Cooled Weak Solution	*60*	*℃*	*52*	*℃*
Inlet Temperature of Sub cooled Strong Solution	*45*	*℃*	*37*	*℃*
Exit Temperature of Preheated Strong Solution	*102.2*	*℃*	*88.45*	*℃*
Mass Flow Rate of Strong solution	*0.08268*	*kg/sec*	*0.03189*	*kg/sec*
Mass Flow Rate of Weak solution	*0.07818*	*kg/sec*	*0.0275*	*kg/sec*
Inlet Enthalpy of Saturated Weak Solution	*325.2*	*kJ/kg*	*290.4*	*kJ/kg*
Exit Enthalpy of Sub-Cooled Weak Solution	*53.64*	*kJ/kg*	*22.33*	*kJ/kg*
Inlet Enthalpy of Sub cooled Strong Solution	*-22.66*	*kJ/kg*	*-67.98*	*kJ/kg*
Exit Enthalpy of Preheated Strong Solution	*234.1*	*kJ/kg*	*163.2*	*kJ/kg*
Heat Duty of the LLHE	*21.23*	*kW*	*7.371*	*kW*

Table 4 . 51 : Thermodynamic Analysis for Pump of Alternative D3

Pump (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Inlet Pressure	*3.011*	*bar*	*3.011*	*bar*
Exit Pressure	*17.69*	*bar*	*14.19*	*bar*
Operating Temperature	*45*	*℃*	*37*	*℃*
Operating Concentration	*0.3777*	*-*	*0.4226*	*-*
Operating Mass Flow Rate	*0.08268*	*kg/sec*	*0.03189*	*kg/sec*
Inlet Enthalpy	*-23.88*	*kJ/kg*	*-68.94*	*kJ/kg*
Exit Enthalpy	*-22.66*	*kJ/kg*	*-67.98*	*kJ/kg*
Inlet Entropy	*0.5366*	*kJ/kgK*	*0.4081*	*kJ/kgK*
Exit Entropy	*0.5351*	*kJ/kgK*	*0.4069*	*kJ/kgK*
Inlet Specific Volume	*0.00117*	*m^3/kg*	*0.00118*	*m^3/kg*
Exit Specific Volume	*0.00117*	*m^3/kg*	*0.00118*	*m^3/kg*
Heat Duty of the Pump (Electrical Energy Required)	*0.1005*	*kW*	*0.03065*	*kW*

Table 4 . 52 : Thermodynamic Analysis for Absorber of Alternative D3

Absorber (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Operating Pressure	*3.011*	*bar*	*3.011*	*bar*
Inlet Temperature of Weak solution	*60*	*℃*	*52*	*℃*
Exit Temperature of Strong solution	*45*	*℃*	*37*	*℃*
Absorption Starting Temperature	*51.75*	*℃*	*53.88*	*℃*
Inlet Temperature of Ammonia Vapors	*18*	*℃*	*14*	*℃*
Inlet Ammonia Mass Concentration of Weak solution	*0.3421*	*-*	*0.3311*	*-*
Exit Ammonia Mass Concentration of Strong solution	*0.3777*	*-*	*0.4226*	*-*
Inlet Mass Flow Rate of Weak solution	*0.07818*	*kg/sec*	*0.0275*	*kg/sec*
Exit Mass Flow Rate of Strong solution	*0.08268*	*kg/sec*	*0.03189*	*kg/sec*
Refrigerant Mass Flow Rate	*0.0045*	*kg/sec*	*0.00438*	*kg/sec*
Inlet Enthalpy of Weak solution	*53.64*	*kJ/kg*	*22.33*	*kJ/kg*
Exit Enthalpy of Strong solution	*-23.88*	*kJ/kg*	*-68.94*	*kJ/kg*
Absorption Starting Enthalpy	*16.84*	*kJ/kg*	*29.98*	*kJ/kg*
Inlet Enthalpy of Ammonia Vapors	*1322*	*kJ/kg*	*1313*	*kJ/kg*
Exit Entropy of Strong solution	*0.5366*	*kJ/kgK*	*0.4081*	*kJ/kgK*
Absorption Starting Entropy	*0.645*	*kJ/kgK*	*0.6787*	*kJ/kgK*
Inlet Entropy of Ammonia Vapors	*4.989*	*kJ/kgK*	*4.956*	*kJ/kgK*
Exit Specific Volume of Strong solution	*0.00117*	*m³/kg*	*0.00118*	*m³/kg*
Absorption Starting Specific Volume	*0.00116*	*m³/kg*	*0.00116*	*m³/kg*
Inlet Specific Volume of Ammonia Vapors	*0.4546*	*m³/kg*	*0.4473*	*m³/kg*
Heat Duty of the Absorber	*12.12*	*kW*	*8.568*	*kW*

Table 4 . 53 : Thermodynamic Analysis for Coolant HX of Alternative D3

Coolant Heat Exchanger (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
Heat Duty of the Coolant Heat Exchanger	*18.395*	*kW*	*14.5708*	*kW*

Table 4 . 54 : Coefficient of Performance of Alternative D3

Coefficient of Performance (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*Yes*	
COP	*0.372*		*0.5218*	
COP Overall	*0.4477*			

Table 4 . 55 : Heat Storage Requirement of Alternative D3

Heat STORAGE (D3.)				
	Day time		*Night time*	
parameter	*Value*	*unit*	*Value*	*unit*
Day Time Operation	*Yes*			
Night Time Operation			*No*	
Storage Pressure	*12*	*bar*		*bar*
Storage Temperature	*152*	*℃*		*℃*
Storage Capacity Required (Heat)	*2815*	*kg*		*kg*

4.12 Development of New Ice Storage Unit

Cold storage systems (D1) requires the use of cold storage units to store cold during the period when cooling effect is produced at the evaporator of the absorption system and utilize this stored cold during the period when refrigeration system is not producing any cooling effect. Hence, cold storage units also play a significant role in a continuously supplying cooling load system.

Andrepont and Steinmann [53] recommended the selection of chilled water as the storage medium for cooling as it requires a relatively low refrigeration capital cost

compared to ice storage-based systems. Cross et al. [54] recommended the use of a hybrid ice-chilled-water thermal storage system by using a chilled water system to meet the larger share of the inlet air-cooling load, and a smaller ice storage system to meet the 4℃ requirements. They concluded that ice storage-based systems and hybrid systems could yield an 11% greater increase in plant capacity compared to chilled water-based systems.

S. M. Hasnain [55] described the inherent pros and cons of the two common (i.e. chilled water and ice storage) commercially available thermal energy storage (TES) technologies for off-peak air conditioning applications. He demonstrated the case studies on cool thermal storage have not only savings in energy and other operation and maintenance costs but also significant savings in initial capital costs. The use of cool thermal storage equipment for gas turbine inlet air-cooling was also examined. The following advantages of ice thermal storage systems over chilled water storage systems were listed.

- *Larger cooling capacity for a given storage volume*
- *Less space requirement, making it attractive for retrofit applications and also for use in new construction*
- *Less thermal losses to the surrounding environment, owing to smaller surface area*
- *Fewer design restrictions, for example, elimination of stratification requirement within the storage tank*
- *Greater reliability due to the availability of packaged systems which usually carry manufacturer's warranty*

- *Accurate estimation of the storage cost during the design phase of the project (due to availability of packaged systems)*
- *Lower cost of maintenance and water treatment (because of packaged design and less circulating water)*
- *Lower storage temperature, reducing the cost of pumping and air distribution (consequently, downsizing pipes, ducts, pumps, etc.,) and*
- *Modularity of storage tank, permitting the use of factory-built tanks*

Although ice storage systems presents a remarkable advantage because of the fact that the latent heat of fusion of ice is very high i.e. 334 kJ/kg compared to the specific heat chilled water i.e. 4.12 kJ/kg-K, yet there are some limitations of ice storage systems that increase the cost of such systems. As we know that ice forms on the cooling surface, this causes the ice to stick to the cooling surface. As ice also behaves as a heat insulator, so increasing the thickness of ice on the cooling surface increases the heat transfer resistance. Thus after a minor period, the ice formations process actually stops. Hence, it is important to remove ice from the cooling surface continuously or at regular intervals.

Removing the ice from the cooling surface can be done either mechanically or by defrosting process. Defrosting processes involves usage of hot defrosting gas to remove the ice from the cooling surface; however, this causes a loss in refrigeration effect to about 10 percent in addition to the complexity involved in operating such a system. The present study introduces a new ice storage system, which is simpler in design and uses mechanical means to remove ice from the cooling surface instead of defrosting system. Following considerations have been taken into account while designing this new ice storage system.

- Ice formation surface should be smooth so that ice can be easily removed out of it.
- Ice formation surface should be at the bottom of ice storage system, so that utilizing density difference of ice from water in order to separate the ice from the cooling surface.
- Ice should be removed from the cooling surface before it becomes thick enough to be easily removed.

Figure 4.34 shows the new ice storage system. This system uses a rotating scrubber to remove ice from the cooling surface. The speed of the rotating scrubber is adjusted such that ice is removed from the cooling surface before it becomes thick enough to be easily removed. Brine solution takes the cold from the evaporator and then moves to the bottom of the circular flat plate. The brine gives the cold to the water in the ice storage system thus producing ice at the cooling surface. The rotating scrubber removes this ice. The top view of rotating scrubber is shown in Figure 4.35.

As rotating scrubber removes the ice from the cooling surface, it follows its way up to the top of ice storage system because the density of ice is less compared to that of water. Similarly, as the brine gives the cold to the ice storage system, it then follows its way up to the evaporator through the thermo siphon effect. The heat transfer area for this ice storage system is actually equal to the sweep area for the rotating scrubber. As the rotating scrubber is only meant to produce the shearing effect to remove ice from the cooling surface, therefore, the fins of the rotating scrubber are actually made almost flat to minimize the amount of water displaced from its rotation thus minimizing the amount of energy required for its rotation.

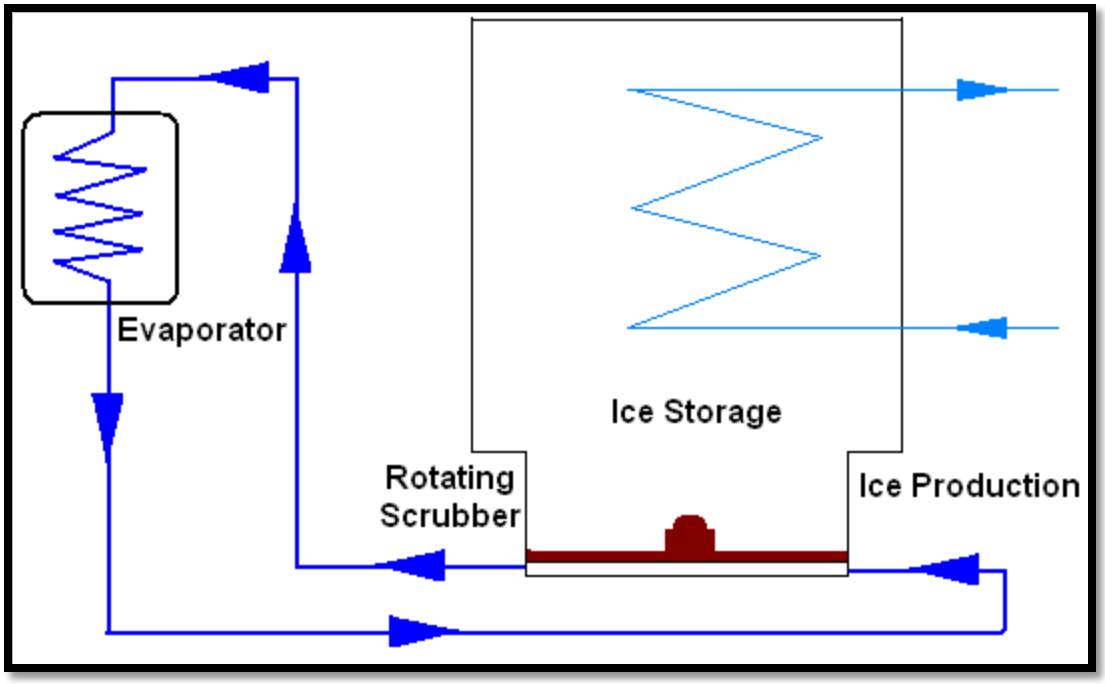

Figure 4 . 34 : New Ice Storage system

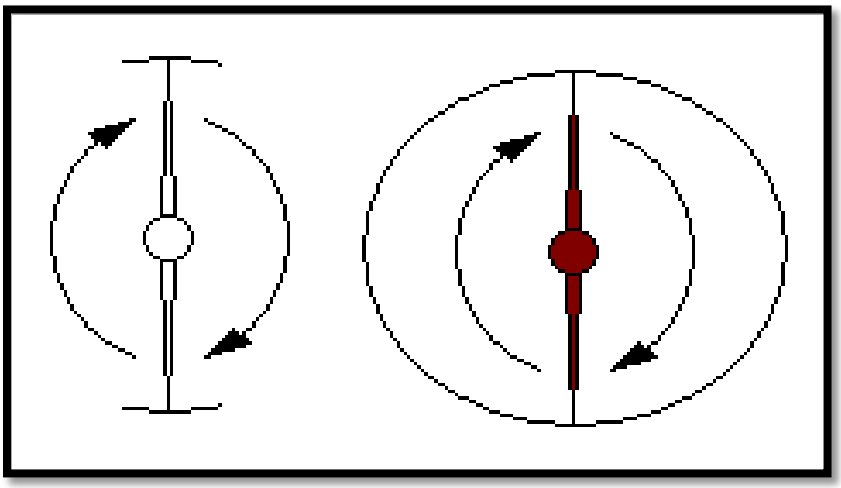

Figure 4 . 35 : Top view of Rotating Scrubber

The ice adhesion shear stress is minimum for stainless steel i.e. 520 kPa [56], thus stainless steel is recommended for the design of cooling surface of this ice storage system. Considering this ice adhesion shear stress and for an ice storage unit with a sweep area of diameter corresponding to the heat duty of the evaporator, the torque required to rotate the scrubber is determined. The speed of the rotating scrubber is determined by assuming thickness of ice to be 5mm at the point when it is removed from the cooling surface. Finally, the energy requirements for all the previously designed cold storage alternatives are detailed in Table 4.56.

Table 4 . 56 : Energy Requirement for Ice Storage Unit

Ice storage system					
Alternative	Mass (kg) Produced	Mass (kg) Stored	Hours	Heat duty (kW)	Electrical power (kW)
D1.1.1	*1295*	*755.4*	*10*	*12*	*4.078*
D1.1.2	*1295*	*755.4*	*10*	*12*	*4.078*
D1.2	*1295*	*937.3*	*6.629*	*18.1*	*6.152*

4.13 Solar Collector Analysis for Alternate Designs

The solar collector analysis is based on the seasonal optimum angle of inclination for Dhahran [57]. Using the developed method for the determination of solar collector sizing [58], the collector area required for the operation is determined for all the alternative designs.

4.13.1 Analysis Using Single Glazed Flat Plate Solar Collector

Table 4.57 shows a single glazed flat plate solar collector with the following specifications.

Table 4 . 57 : Specifications of Single Glazed Flat Plate Solar Collector

Gross Area	*23.1 ft2*
Aperture Area	*21.5 ft2*
Absorption / Emission	*95% / 5%*
Transmission of Covering	*88%*
Heat Loss Coefficient – a1	*(k1) 3.95 W/(m2K)*
Heat Loss Coefficient – a2	*(k2) 0.0165 W/(m2K)*

Since single glaze flat plate solar collectors are used at comparatively low generator temperatures therefore, their utilization is analyzed for double stage systems only, operating at about 50 ℃ higher than the ambient throughout the year for Dhahran using seasonal optimum angle of inclinations. From the thermodynamic analysis of double stage systems, the heat duty determined at the generators are utilized to determine the size of the single glazed flat plate solar collector required for the operation given in Table 4.58.

Table 4 . 58 : Single Glazed Flat Plate Solar Collector Area Required

Solar collector sizing (single glaze flat plate)					
Alternative	Stage	Heat Duty (kW)	Hours	Daily energy requirement (MJ)	Collector area required (sqm)
D1.1.2	*First*	*57.38*	*10*	*2065.68*	*300*
D1.1.2	*Second*	*18.02*	*10*	*648.72*	*100*
D2.2	*First*	*52.19*	*10*	*1878.84*	*250*
D2.2	*Second*	*18.02*	*10*	*648.72*	*100*

4.13.2 Analysis Using Double Glazed Flat Plate Solar Collector

Table 4.59 shows a double glazed flat plate solar collector with the following specifications.

Table 4 . 59 : Specifications of Double Glazed Flat Plate Solar Collector

Absorber Area	*2.7 m2*
Absorption / Emission	*95% / 5%*
Transmission of Covering	*88%*
Heat Loss Coefficient – a1	*(k1) 2.4 W/(m2K)*
Heat Loss Coefficient – a2	*(k2) 0.015 W/(m2K)*
Operating Temperature	*80 – 150 C*

Since double glaze flat plate solar collectors can be used at comparatively high generator temperatures therefore, their utilization is analyzed for single stage systems operating at about 90 ℃ higher than the ambient throughout the year for Dhahran using seasonal optimum angle of inclinations. From the thermodynamic analysis of different

alternative systems, the heat duty determined at the generators is utilized to determine the size of the double-glazed flat plate solar collector required for the operation as given in Table 4.60.

Table 4 . 60 : Double Glazed Flat Plate Solar Collector Area Required

Solar collector sizing (Double Glaze Flat Plate)					
alternative	Period of Operation	Heat Duty (kW)	Hours	Daily energy requirement (MJ)	Collector area required (sqm)
D1.1.1		*31.92*	*10*	*1149.12*	*300*
D1.2		*50.2*	*10*	*1807.2*	*280*
D2.1		*27.83*	*10*	*1001.88*	*250*
D3	*Day*	*13.3*	*10*	*478.8*	*125*
D3 ($T_{ambient}$ + 130)	*Night*	*9.54*	*14*	*480.816*	*Not Possible*

4.13.3 Analysis Using Evacuated Tubular Solar Collector

Table 4.61 shows an evacuated tubular solar collector with the following specifications.

Table 4 . 61 : Specifications of Evacuated Tubular Solar Collector

Heat Loss: (a1)	*1.529W/m2 K (.269 (Btu/h)/ft2/F)*
Heat Loss: (a2)	*0.0166W/m2 K2 (.003 (Btu/h)/ft2/F)*
Transmission Through Glass	*>92% (AM1.5)*
Absorptivity Coefficient:	*> 94%*
Emittance	*< 7%*

Since evacuated tubular solar collectors can be used at comparatively high generator temperatures therefore, their utilization is analyzed for single stage systems operating at about 90 ℃ higher than the ambient throughout the year for Dhahran using seasonal optimum angle of inclinations. From the thermodynamic analysis of different

alternative systems, the heat duty determined at the generators is utilized to determine the size of the evacuated tubular solar collector required for the operation as given in Table 4.62.

Table 4 . 62 : Evacuated Tubular Solar Collector Area Required

Solar collector sizing (Evacuated Tubular)					
Alternative	Period of Operation	Heat Duty (kW)	Hours	Daily energy requirement (MJ)	Collector area required (sqm)
D1.1.1		*31.92*	*10*	*1149.12*	*160*
D1.2		*50.2*	*10*	*1807.2*	*180*
D2.1		*27.83*	*10*	*1001.88*	*135*
D3	*Day*	*13.3*	*10*	*478.8*	*70*
D3 ($T_{ambient}$ + 130)	*Night*	*9.54*	*14*	*480.816*	*Not Possible*

CHAPTER 5

RESULTS AND DISCUSSIONS

The cost of any absorption system depends upon the cost of its individual components. The cost of individual components further depends upon the heat duty of the components. Higher heat duty corresponds to higher heat exchange area required for that component which further corresponds to the higher cost of that component. After undergoing a detailed thermodynamic analysis for all the design alternatives for solar powered aqua-ammonia refrigeration system, this section undergo a detailed comparative analysis for all the alternative design, which will form the basis of selection of optimum design.

Figure 5.1 shows the bar chart for generator heat duty of all the analyzed alternative designs. It can be seen from the Figure 5.1 that the generator heat duty for D3 (heat storage system) is the lowest and for the alternatives D1.1.2 and D2.2 (double stage systems) are the highest. Similarly, Figure 5.2 shows the bar chart for dephlegmator heat duty of all the analyzed alternative designs. It can be seen from the Figure 5.2 that the dephlegmator heat duty for the intermittent system without rectification is the highest.

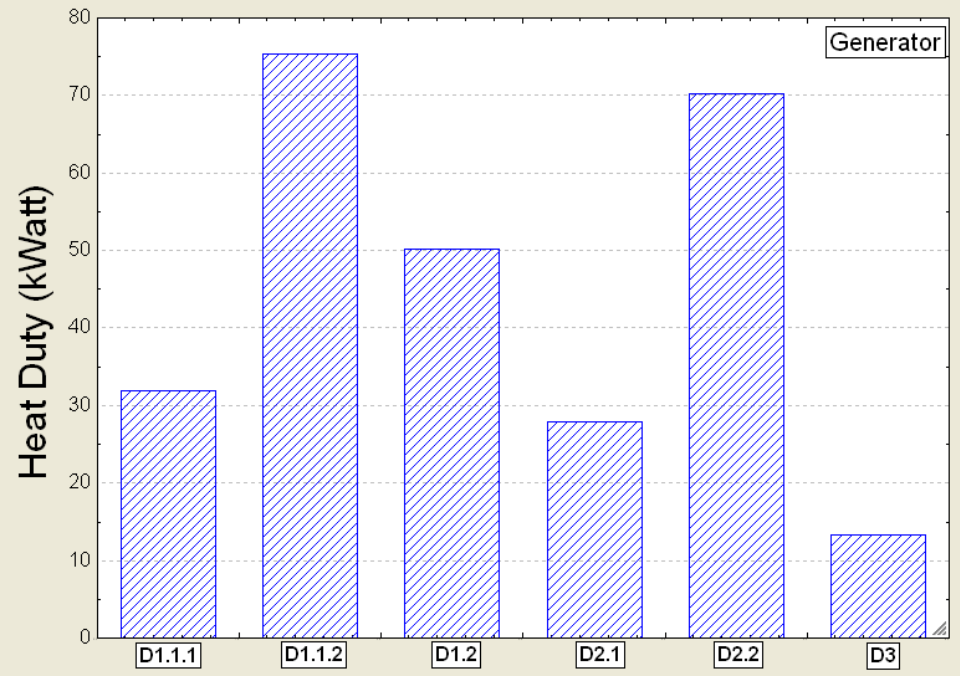

Figure 5 . 1 : Bar Chart for Generator Heat Duty

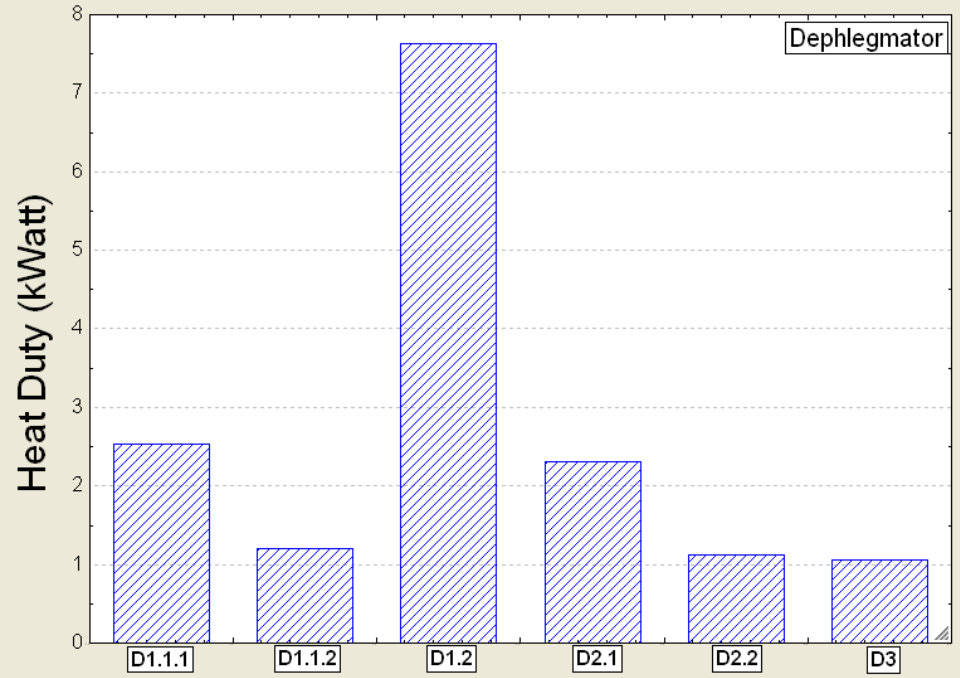

Figure 5 . 2 : Bar Chart for Dephlegmator Heat Duty

The reason for very high generator heat duty in case of double stage systems is that, in such systems, there are two generators that require heat input for the operation. However, in case of heat storage system D3, the generator is in operation for 24 hours so the total heat load is divided over 24 hours, thus the generator heat duty is low for such systems.

Figure 5.3 shows the bar chart for condenser heat duty of all the analyzed alternative designs. Since condenser heat duty is directly proportional to generator heat duty, therefore, it can be seen from the Figure 5.3 that the generator heat duty for D3 (heat storage system) is the lowest and for the alternatives D1.1.2 and D2.2 (double stage systems) are the highest.

Figure 5.4 shows the bar chart for VLHE heat duty of all the analyzed alternative designs. It can be seen that the VLHE heat duty for the cold storage systems is generally high compared to refrigerant storage or heat storage systems

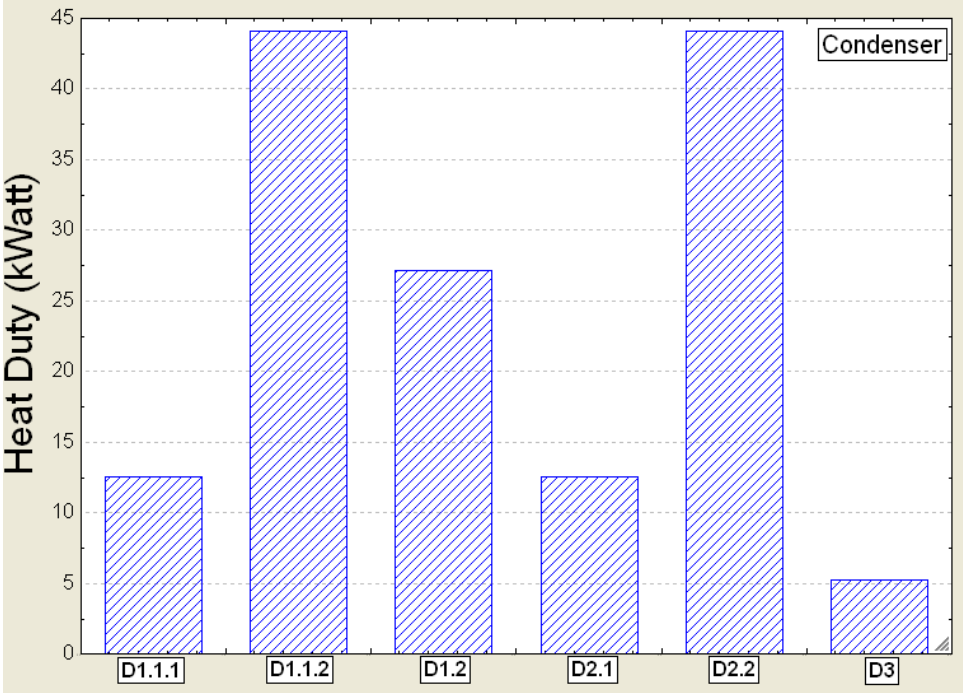

Figure 5 . 3 : Bar Chart for Condenser Heat Duty

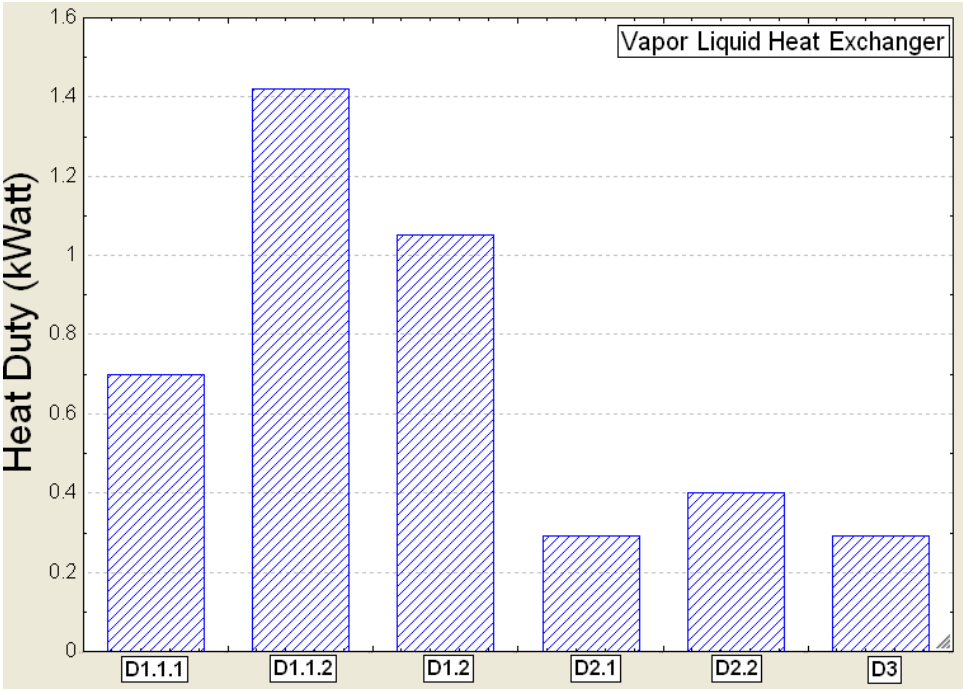

Figure 5 . 4 : Bar Chart for VLHE Heat Duty

The reason for very high VLHE heat duty in case of cold storage systems is that, in such systems, as the absorption process is limited to day time or night time operation only, so the refrigerant mass flow rate is high in order to produce the required refrigeration effect. However, in case of refrigerant or heat storage system, the absorption process is spread for 24 hours. As heat duty of the VLHE is a direct function of refrigerant mass flow rate, thus the VLHE heat duty is low for such systems.

Figure 5.5 shows the bar chart for Evaporator heat duty of all the analyzed alternative designs. It can be seen that the Evaporator heat duty for the cold storage systems is generally high compared to refrigerant storage or heat storage systems. The highest heat duty of the evaporator is for cold storage double stage system and the lowest heat duty is for heat storage and for single stage refrigerant storage systems.

Figure 5.6 shows the bar chart for LLHE heat duty of all the analyzed alternative designs. It can be seen from the Figure 5.6 that the LLHE heat duty for the alternatives D1.1.2 and D2.2 (double stage systems) are the highest because of high aqua-ammonia mass flow rates in the first stage of such cycles.

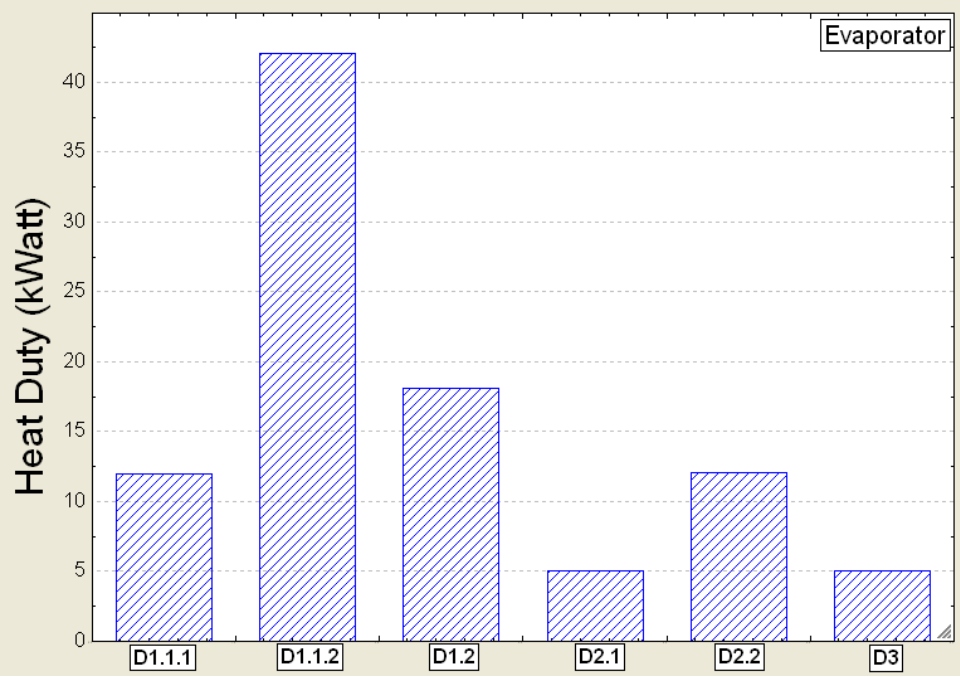

Figure 5 . 5 : Bar Chart for Evaporator Heat Duty

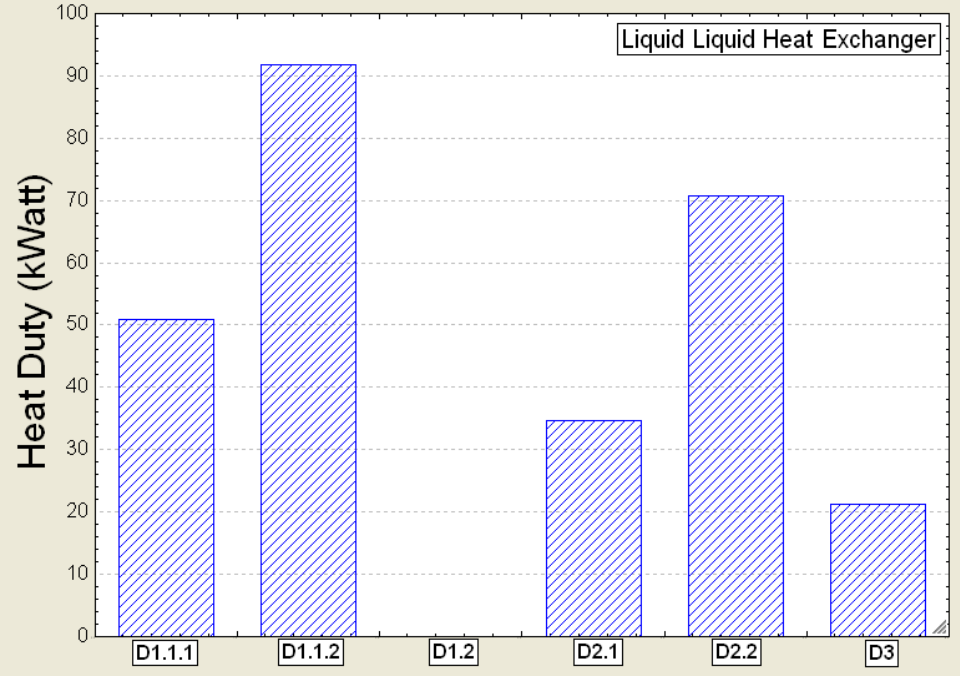

Figure 5 . 6 : Bar Chart for LLHE Heat Duty

Figure 5.7 shows the bar chart for Absorber heat duty of all the analyzed alternative designs. It can be seen that the Absorber heat duty for the cold storage systems is generally high compared to refrigerant storage or heat storage systems. The highest heat duty of the absorber is for cold storage double stage system.

Figure 5.8 shows the bar chart for coolant heat exchanger heat duty of all the analyzed alternative designs. It can be seen that the coolant heat exchanger heat duty for the double stage systems is generally high. The reason being high heat rejection at the absorber and condenser of such systems.

Figure 5.9 shows the bar chart for collector size of all the alternative designs. From the Figure 5.9, it can be seen that although the double stage systems utilized single glazed flat plate solar collector, which is cheaper than the evacuated tubular solar collector is, yet because of very low COP the collector area required for the operation becomes so high that it increases the overall cost of the system considerably. Furthermore it can be seen from the Figure 5.9 that heat storage systems requires the use of concentrators because the temperature required for the storing heat so high that even it cannot be collected from evacuated tubular solar collector. Along with that, it can also be seen that refrigerant storage systems requires reasonably small collector area for operation.

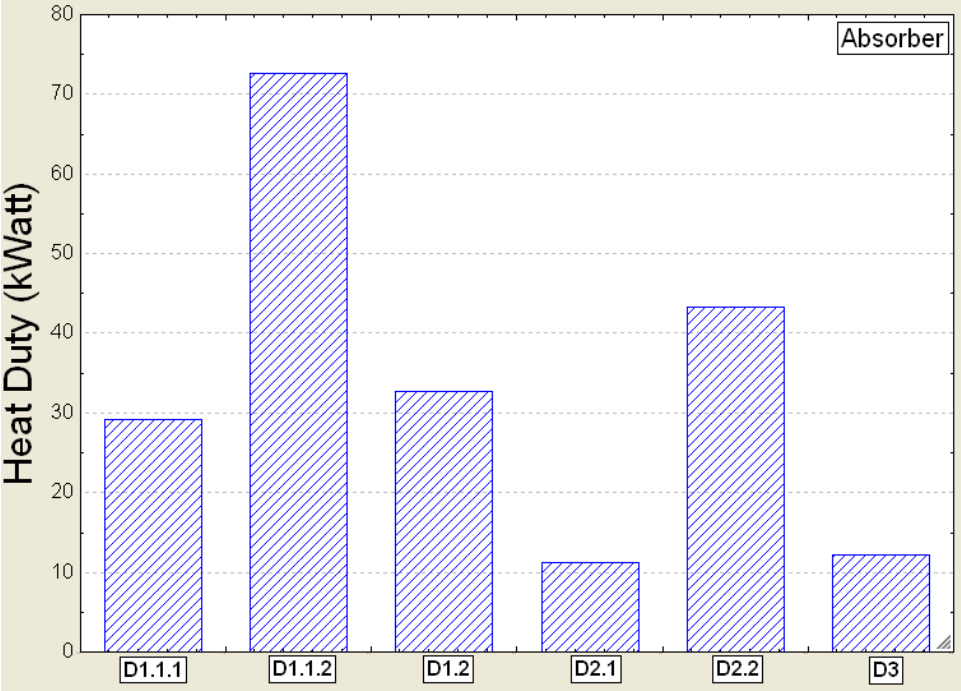

Figure 5 . 7 : Bar Chart for Absorber Heat Duty

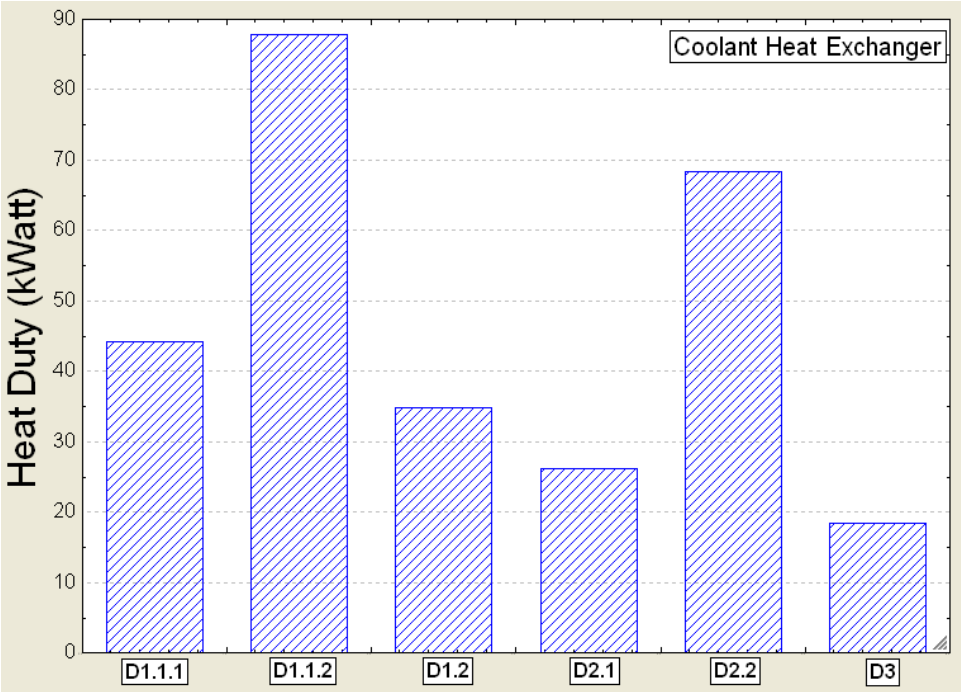

Figure 5 . 8 : Bar Chart for Coolant HX Heat Duty

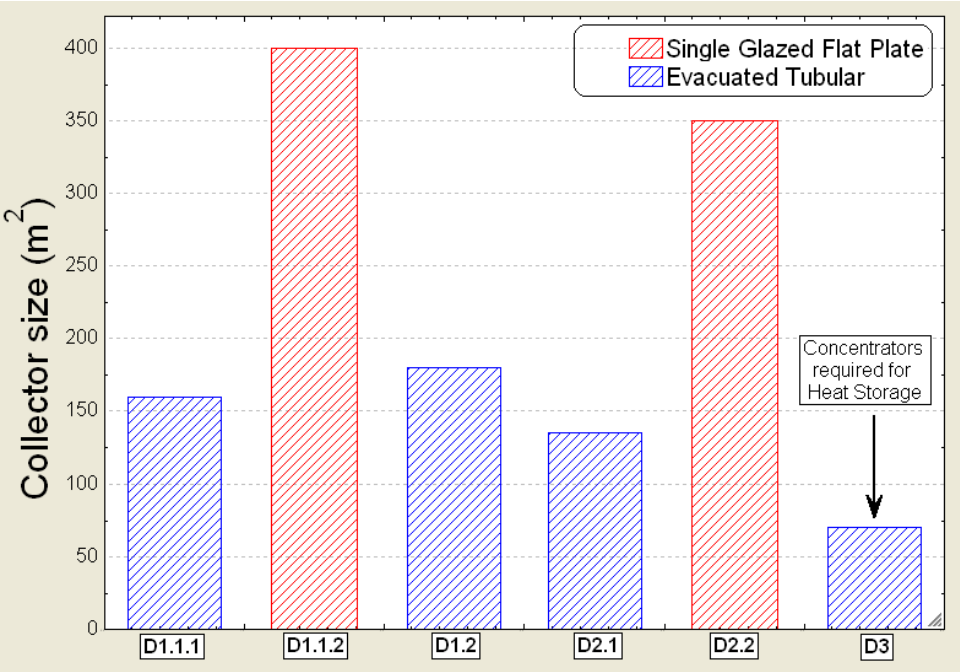

Figure 5 . 9 : Bar Chart for Collector Size

The results for all the systems in terms of performance of the alternatives are shown in the bar chart in Figure 5.10. Similarly, the final results for all the systems in terms of Mass of the system for all the alternatives is shown in Figure 5.11. From the analysis of the bar chart in Figure 5.10, it shows that the Alternative D3 presents the highest COP for the system, however the system mass as heat storage turns out to be quite high i.e. 2815 kg. As this amount needs to be stored at around 152 – 160 ℃, a lot of heavy insulation will be required to store this amount of heat which will increase the initial cost of the system.

Also it can be seen from chapter 4 that none of the single glaze, double glaze flat plate or evacuated tubular collector will be able to produce that high temperature in the climate of Dhahran, hence concentrators will be required to produce that amount of heat storage which will also significantly increase the cost of the system. Hence Alternative D3 turns out to be unsuitable for the operation.

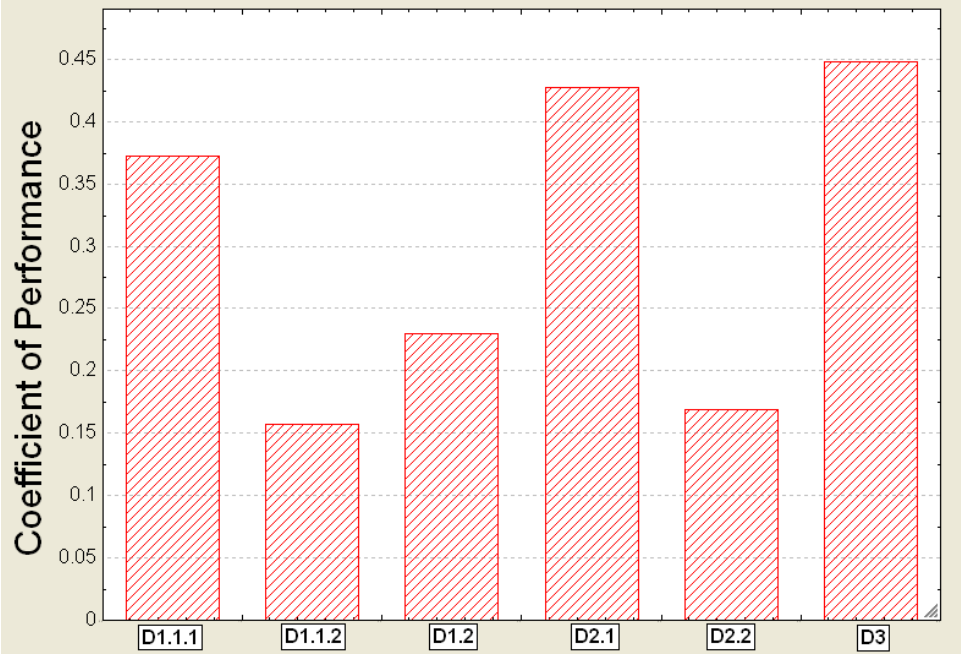

Figure 5 . 10 : Coefficient of Performance Bar Chart for all the alternatives

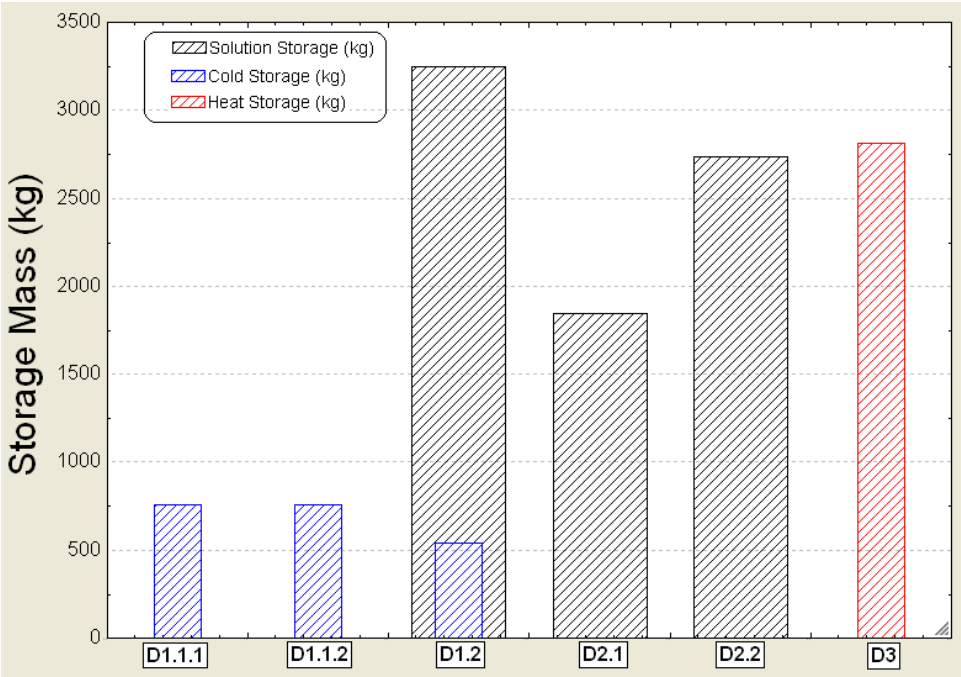

Figure 5 . 11 : System Mass Bar Chart for all the alternatives

Alternatives D1.1.2 and D2.2 have very low coefficient of performance which makes the system unfeasible for the operation. Alternative D2.1 turn out to be the best system in terms of performance of the system as seen from figure 5.10. Also the system mass for alternative D2.1 is quite reasonable in size, simple in design and also do not require any insulation. Hence the Alternative D2.1 turns out to be the best system in operation. Although the storage mass for the alternative D1.1.1 is even lower than the Alternative D2.1, yet due to complex design of cold storage and comparatively low COP makes it the second best alternative for the operation. Finally, although the COP of alternative D1.2 is very less and mass of this alternative is comparatively very high, yet the simplicity in the design of the system makes it third best alternative for the operation.

Cost Analysis of the Selected Alternative Designs

Along with the COP of the system and the system's mass, the other important parameter involved in the selection of the most suitable design is the cost analysis. For estimating the cost of a system, it is required to estimate the individual cost of each and every component of the system. The cost of each and every component of the system depends upon

- The heat rating of the component
- Specific operation performed by that component and
- The type of heat exchange fluid involved

Verla et al [11] has provided the heat rating of each component and estimated the cost of each component utilized in their design. This provides us a basis of determining the cost per unit UA for each component in our analysis. Knowing the estimated cost per

UA value for each component utilized, the actual cost of the selected alternatives can be effectively estimated. Similarly for storage tanks, the cost per unit kg of the storage is used to estimate the actual cost of the storage in consideration.

The cost of the solar collection system is determined by the cost per unit aperture area of the solar collector module as provided by the manufacturer which is 500$/sq.meter for double glazed flat plate solar collector and 551.21$/sq.meter for evacuated tubular solar collector. Similarly, the cost of the photovoltaic system is based on the cost per unit watt of solar generated electricity which is equal to 1.8$/watt. This solar generated electricity is mainly required by the pumps and the rotating scrubber of the ice storage system.

To determine the energy total that must be provided by the photovoltaic system, it is also necessary to consider the energy that the components of the photovoltaic system consume. This self-consumption energy can be calculated with the energy efficiency of the elements of the system. The values are as follows: current invertor 95%, charge controller 95%, lead acid batteries 90%. Hence, the estimated cost of some of the alternatives is shown in Table 5.1.

Table 5 . 1 : Cost Estimation for Selected Alternative Designs

	ALTERNATIVES		
EQUIPMENT	D2.1	D1.1.1	D1.2
Generator	*5880.39*	*7258.04*	*4476.50*
Rectifier	*467.72*	*473.48*	*-*
Dephlegmator	*390.92*	*417.19*	*1211.61*
Condenser	*2647.01*	*2647.01*	*-*
Refrigerant Storage Tank	*2119.34*	*-*	*3633.69*
Reflux Tank	*-*	*-*	*98.58*
Vapor Liquid Heat Exchanger	*1383.05*	*3319.33*	*4919.11*
Expansion Device	*135.00*	*135.00*	*135.00*
Expansion Device	*65.00*	*65.00*	*-*
Evaporator	*5867.17*	*14081.20*	*21239.14*
Liquid-Liquid Heat Exchanger	*7766.95*	*12444.71*	*-*
Weak Solution Storage	*3766.70*	*-*	*-*
Absorber	*6981.03*	*16318.49*	*10240.61*
Strong Solution Storage	*4289.63*	*-*	*7553.93*
Liquid Ammonia	*500.00*	*75.00*	*750.00*
Pump	*375.00*	*375.00*	*-*
Coolant Heat Exchanger	*21111.01*	*35686.04*	*28081.75*
Ice Storage Unit	*-*	*3000.00*	*5000.00*
Photovoltaic System	*189.63*	*9571.86*	*9037.86*
~~*Solar Collection System (Double Glazed)*~~	~~*125000.00*~~	~~*150000.00*~~	~~*140000.00*~~
Solar Collection System (Evacuated Tubular)	*74413.35*	*88193.60*	*99217.80*
TOTAL ESTIMATED COST	138348.9	194060	1941357

From the cost analysis, it can be seen that although the double glazed solar collectors are comparatively cheaper than evacuated tubular solar yet the net cost by evacuated tubular collectors are comparatively much less because of less surface area required. It can be seen that the cost of alternative D2.1 is much less compared to the cost of D1.1.1 and D1.2. As the heat ratings of the LLHE, coolant heat exchanger, absorber and evaporator are much high in D1.1.1 compared to D2.1 therefore it increases the cost

of D1.1.1 compared to D2.1. The other important parameter is the cost of the photovoltaic system. As cold storage unit is utilized in the alternative D1.1.1 and D1.2 which requires considerably high electrical energy for the operation, this increase the cost of photovoltaic system for these alternatives compared to D2.1. Similarly for the alternative D1.2, although system requires less components for the operation yet because of the low COP of the system, the cost of the solar collection system is above the cost of others. Also because of high heat rating for evaporator, absorber and generator, this further increases the cost of the alternative D1.2. Finally from the COP analysis, system's size analysis and cost analysis, it is determined that newly suggested alternative design D2.1, presents the most simple, suitable and economical design for totally solar powered aqua-ammonia vapor absorption refrigeration system for continuous operation.

The performance curves of the selected optimum alternative design for aqua-ammonia vapor absorption refrigeration system are shown in Figure 5.12 and Figure 5.13.

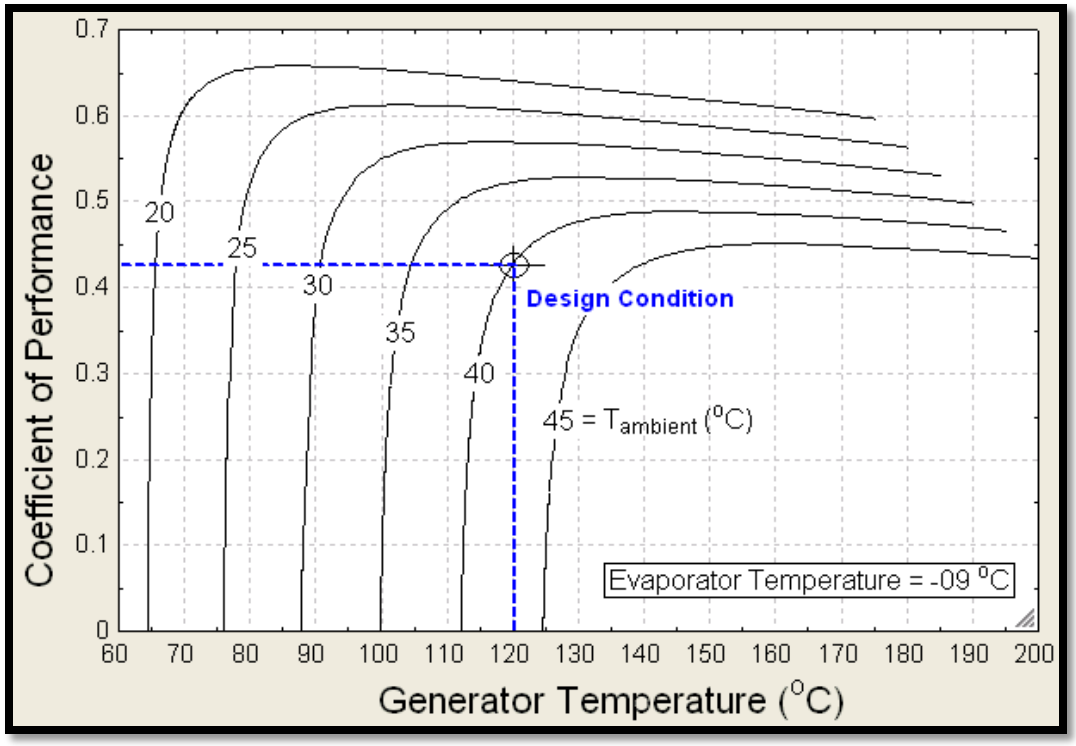

Figure 5 . 12 : Performance Curves against varying Generator Temperature

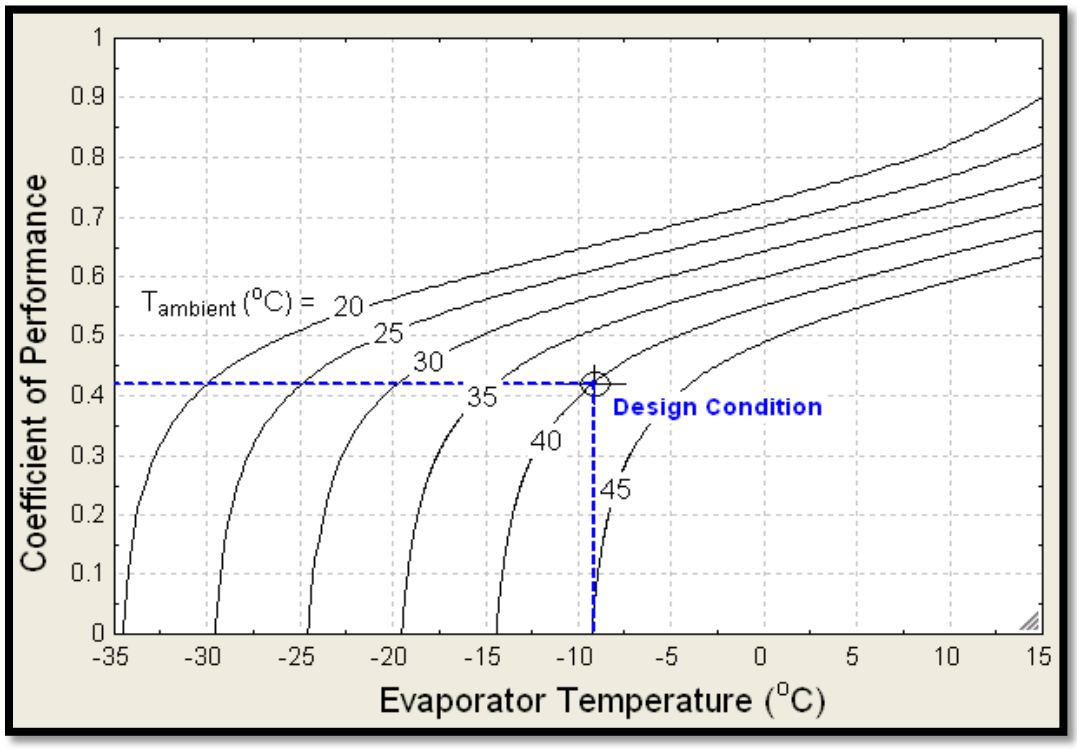

Figure 5 . 13 : Performance Curves against varying Evaporator Temperature

CHAPTER 6

CONCLUSION AND RECOMMENDATION

In this thesis work, several alternative designs based on solar powered aqua-ammonia vapor absorption refrigeration systems for continuous cooling requirements were suggested. The suggested designs were broadly categorized as cold storage, refrigerant storage and heat storage systems.

The refrigerant storage system based on continuous operation was determined to be best-suited design for the purpose. It is because of the fact that the COP of the refrigerant storage system based on continuous operation turns out to be even higher than the conventional continuously operating vapor absorption system while the system mass required for 24 hours operation is much less than the system mass required for conventional intermittently operating vapor absorption system.

Although the heat duties of different component of the heat storage systems is found relatively low compared to other alternatives, as well as the COP of the heat storage system was also found high comparatively, yet it was found that heat storage systems are not suited for operation. It is because of the fact that none of the double glazed flat plate or evacuated tubular solar collector are able to provide heat for storage as the temperature required for heat storage is very high compared to others. This requires the use of concentrators for the operation along with the solar tracking system

for the concentrators. Along with that, the mass required for the heat storage requires heavy insulation because it will be at a very high temperature compared to that of the ambient. Hence, the overall cost of the system will increase considerably.

It was found that because of the compact design and high COP, the continuous based systems are more feasible than the intermittent based systems. The visible advantage in the intermittent systems is that they require fewer components for operation; however, it was found that because of low COP, this visible advantage is also nullified by the additional cost of high collector area required for the operation. Along with that, the heat duties of different component of intermittent systems are also considerably high compared to those of the continuous systems, which further add fuel to the fire.

Along with that, in this thesis, continuous operation based double stage systems were also compared with single stage systems. It was found that single stage systems are far better in performance compared to double stage systems. It is because of the fact that double stage systems, having very low COP, require considerably high collector area for the operation. Along with that, the heat duties of different components of double stage systems are also much higher than those of single stage systems. The only visible advantage in case of double stage systems is that, they can be operated at relatively very low generator temperature thus requires only cheap single glazed flat plate collector. But this visible advantage is nullified by the fact that the collector area required for double stage systems is much higher than that of single stage systems which makes the overall cost of the double stage systems to becomes much higher than that of the single stage systems.

Along with that, in this thesis, double glazed flat plate was also compared with evacuated tubular solar collectors. It was found that although the cost of evacuated tubular solar collectors is quite high compared to double glazed flat plate solar collectors, yet the overall cost of the evacuated tubular solar collector system is lower than double glazed flat plate solar collector system because of very low collector area requirement for evacuated tubular solar collector. Hence, the evacuated tubular solar collector system is found to be more feasible for the operation.

Along with that, in this thesis, a new ice storage system is suggested for the cold storage systems. The suggested design is simple and based on the mechanical means of ice removal from the cooling surface by the help of rotating scrubber located at the bottom of the ice storage unit.

Since the research in solar powered aqua-ammonia vapor absorption refrigeration system is ongoing, it is recommended to undergo a detailed analysis for heat storage systems. Although the heat storage system, require concentrators for the operation, yet the heat duties of different components of the system is determined to be quite low as well as the COP for the system is also determined to be high. It is also recommended to make use of PCM (phase change materials) which may also considerably reduce the size of the heat storage unit. In future, the development of an experimental prototype based on the outcome of this study, for a 24 hours operating solar powered aqua ammonia absorption refrigeration system is also recommended using refrigerant storage system.

APPENDICES

APPENDIX A1 (EES Module for Alternative Design D1.1.1)

```
" Units deg C, bar, kJ, kg, m, sec, kW "
" sc  = solar collector  "
" ********************************  Input Data
********************************************** "
T_gen = 120
T_cond = 45
T_evap = -9
T_abs = 45
Conc_deph_out = 0.996
eta_LLHX = 0.8
eta_VLHX = 0.5
eta_Rectifier = 0.75
Refrigeration_Effect_D&N = 5
Hours_Day = 10
" ******************************** SOLUTION
**************************************************"
" Required Refrigeration Effect"
Refrigeration_Effect_Day = Refrigeration_Effect_D&N*24/Hours_Day
ICE_STORAGE = (24 - Hours_Day) * 3600 * Refrigeration_Effect_D&N / Enthalpy_fusion(Water)
" Parameters Selection "
CALL NH3H2O(138,T_evap + 273,1,1: T_evap_out +
273,P_evap_out,x_evap_out,h_evap_out,s_evap_out,u_evap_out,v_evap_out,q_evap_out)
Pressure_low = P_evap_out
CALL NH3H2O(128,T_abs +273,Pressure_low,0: T_abs_out
+273,P_abs_out,x_abs_out,h_abs_out,s_abs_out,u_abs_out,v_abs_out,q_abs_out)
Conc_high = x_abs_out
CALL NH3H2O(138,T_cond + 273,Conc_deph_out,0: T_cond_out +
273,P_cond_out,x_cond_out,h_cond_out,s_cond_out,u_cond_out,v_cond_out,q_cond_out)
Pressure_high = P_cond_out
CALL NH3H2O(128,T_gen + 273,Pressure_high,0: T_gen_liq_out +
273,P_gen_liq_out,x_gen_liq_out,h_gen_liq_out,s_gen_liq_out,u_gen_liq_out,v_gen_liq_out,q_gen_liq_o
ut)
Conc_low = x_gen_liq_out
CALL NH3H2O(238,Pressure_high,Conc_high,0: T_gen_start +
273,P_gen_start,x_gen_start,h_gen_start,s_gen_start,u_gen_start,v_gen_start,q_gen_start)
CALL NH3H2O(238,Pressure_low,Conc_low,0: T_abs_start +
273,P_abs_start,x_abs_start,h_abs_start,s_abs_start,u_abs_start,v_abs_start,q_abs_start)
" Vapor Liquid Heat Exchanger "
eta_VLHX = (T_vap_out_VLHX - T_evap_out)/(T_cond_out - T_evap_out)
CALL NH3H2O(123,T_vap_out_VLHX + 273,Pressure_low,1: T_abs_vap_in +
273,P_abs_vap_in,x_abs_vap_in,h_abs_vap_in,s_abs_vap_in,u_abs_vap_in,v_abs_vap_in,q_abs_vap_in)
h_liq_out_VLHX = h_cond_out - (h_abs_vap_in - h_evap_out)
CALL NH3H2O(234,Pressure_high,Conc_deph_out,h_liq_out_VLHX: T_before_expansion +
273,P_before_expansion,x_before_expansion,h_before_expansion,s_before_expansion,u_before_expansio
n,v_before_expansion,q_before_expansion)
" Ammonia Expansion Process "
```

```
CALL NH3H2O(234,Pressure_low,Conc_deph_out,h_before_expansion: T_after_expansion +
273,P_after_expansion,x_after_expansion,h_after_expansion,s_after_expansion,u_after_expansion,v_after
_expansion,q_after_expansion)
" efrigerant Flow Rate"
Refrigeration_Effect_Day = m_dot_Refrigerant * (h_evap_out - h_after_expansion)
" Energy Exchange at VLHX "
Q_VLHX1 = m_dot_Refrigerant * (h_cond_out - h_liq_out_VLHX)
Q_VLHX2 = m_dot_Refrigerant * (h_abs_vap_in - h_evap_out)
" High and Low Concentration Mass Flow Rates "
m_dot_low_concentration + m_dot_Refrigerant = m_dot_high_concentration
m_dot_low_concentration*Conc_low + m_dot_Refrigerant*Conc_deph_out =
m_dot_high_concentration*Conc_high
" Pump Work "
CALL NH3H2O(123,T_abs_out + 273,Pressure_high,Conc_high: T_pump_out +
273,P_pump_out,x_pump_out,h_pump_out,s_pump_out,u_pump_out,v_pump_out,q_pump_out)
Q_pump_work1 = m_dot_high_concentration * (h_pump_out - h_abs_out)
Q_pump_work2 = m_dot_high_concentration * v_abs_out * (Pressure_high - Pressure_low) * 100
" Liquid Liquid Heat Exchanger "
CALL
LiqLiqHeatExch(Pressure_high,Pressure_low,Conc_high,Conc_low,eta_LLHX,h_pump_out,h_gen_liq_ou
t,m_dot_high_concentration,m_dot_low_concentration,T_pump_out,T_gen_liq_out:Q_LLHX,T_gen_in,h_
gen_in,q_gen_in,T_abs_liq_in,h_abs_liq_in,q_abs_liq_in)
" Absorber Energy Requirements "
Q_absorber = m_dot_low_concentration*h_abs_liq_in + m_dot_Refrigerant*h_abs_vap_in -
m_dot_high_concentration*h_abs_out
" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high,Conc_deph_out,1: T_cond_in +
273,P_cond_in,x_cond_in,h_cond_in,s_cond_in,u_cond_in,v_cond_in,q_cond_in)
" Condenser Energy Requirements "
Q_condenser = m_dot_Refrigerant * (h_cond_in - h_cond_out)
" Generator Exit Conditions "
CALL
GeneratorLiquidHeating(Conc_high,Pressure_high,Conc_low,m_dot_high_concentration:T_gen_vap_out,h
_gen_vap_out,x_gen_vap_out)
" Rectifier Analysis "
eta_Rectifier = (T_gen_vap_out - T_deph_in1)/(T_gen_vap_out - T_gen_in)
" Principal Operating Line "
CALL NH3H2O(128,T_deph_in1 + 273,Pressure_high,1: T_deph_in +
273,P_deph_in,x_deph_in,h_deph_in,s_deph_in,u_deph_in,v_deph_in,q_deph_in)
Slope = (h_deph_in - h_gen_in)/(x_deph_in - Conc_high)
h_deph_in = Slope*x_deph_in + constant
" Generator Energy Requirements "
Pole_Generator = Slope*Conc_low + constant
Q_generator = m_dot_low_concentration * (h_gen_liq_out - Pole_Generator)
" Dephlegmator Energy Requirements "
Pole_Dephlegmator = Slope*Conc_deph_out + constant
Q_dephlegmator = m_dot_Refrigerant * (Pole_Dephlegmator - h_cond_in)
" Coefficient of Performance "
COP = Refrigeration_Effect_Day/(Q_generator + Q_pump_work2)
" Energy Balance "
Energy_Balance_in = Refrigeration_Effect_Day + Q_generator + Q_pump_work1
Energy_Balance_out = Q_condenser + Q_absorber + Q_dephlegmator
```

APPENDIX A2 (EES Module for Alternative Design D1.1.2)

```
" Units deg C, bar, kJ, kg, m, sec, kW "
" sc  = solar collector  "
" ********************************  Input Data
********************************************** "
T_gen = 120
T_cond = 45
T_evap = -9
T_abs = 45
Conc_deph_out = 0.996
eta_LLHX = 0.8
eta_VLHX = 0.5
eta_Rectifier = 0.75
Refrigeration_Effect_D&N = 5
Hours_Day = 10
" ******************************** SOLUTION
**************************************************"
" Required Refrigeration Effect"
Refrigeration_Effect_Day = Refrigeration_Effect_D&N*24/Hours_Day
ICE_STORAGE = (24 - Hours_Day) * 3600 * Refrigeration_Effect_D&N / Enthalpy_fusion(Water)
" Parameters Selection "
CALL NH3H2O(138,T_evap + 273,1,1: T_evap_out +
273,P_evap_out,x_evap_out,h_evap_out,s_evap_out,u_evap_out,v_evap_out,q_evap_out)
Pressure_low = P_evap_out
CALL NH3H2O(128,T_abs +273,Pressure_low,0: T_abs_out
+273,P_abs_out,x_abs_out,h_abs_out,s_abs_out,u_abs_out,v_abs_out,q_abs_out)
Conc_high = x_abs_out
CALL NH3H2O(138,T_cond + 273,Conc_deph_out,0: T_cond_out +
273,P_cond_out,x_cond_out,h_cond_out,s_cond_out,u_cond_out,v_cond_out,q_cond_out)
Pressure_high = P_cond_out
CALL NH3H2O(128,T_gen + 273,Pressure_high,0: T_gen_liq_out +
273,P_gen_liq_out,x_gen_liq_out,h_gen_liq_out,s_gen_liq_out,u_gen_liq_out,v_gen_liq_out,q_gen_liq_o
ut)Conc_low = x_gen_liq_out
CALL NH3H2O(238,Pressure_high,Conc_high,0: T_gen_start +
273,P_gen_start,x_gen_start,h_gen_start,s_gen_start,u_gen_start,v_gen_start,q_gen_start)
CALL NH3H2O(238,Pressure_low,Conc_low,0: T_abs_start +
273,P_abs_start,x_abs_start,h_abs_start,s_abs_start,u_abs_start,v_abs_start,q_abs_start)
" Vapor Liquid Heat Exchanger "
eta_VLHX = (T_vap_out_VLHX - T_evap_out)/(T_cond_out - T_evap_out)
CALL NH3H2O(123,T_vap_out_VLHX + 273,Pressure_low,1: T_abs_vap_in +
273,P_abs_vap_in,x_abs_vap_in,h_abs_vap_in,s_abs_vap_in,u_abs_vap_in,v_abs_vap_in,q_abs_vap_in)
h_liq_out_VLHX = h_cond_out - (h_abs_vap_in - h_evap_out)
CALL NH3H2O(234,Pressure_high,Conc_deph_out,h_liq_out_VLHX: T_before_expansion +
273,P_before_expansion,x_before_expansion,h_before_expansion,s_before_expansion,u_before_expansio
n,v_before_expansion,q_before_expansion)
" Ammonia Expansion Process "
CALL NH3H2O(234,Pressure_low,Conc_deph_out,h_before_expansion: T_after_expansion +
273,P_after_expansion,x_after_expansion,h_after_expansion,s_after_expansion,u_after_expansion,v_after
_expansion,q_after_expansion)
" Refrigerant Flow Rate"
Refrigeration_Effect_Day = m_dot_Refrigerant * (h_evap_out - h_after_expansion)
" Energy Exchange at VLHX "
Q_VLHX1 = m_dot_Refrigerant * (h_cond_out - h_liq_out_VLHX)
Q_VLHX2 = m_dot_Refrigerant * (h_abs_vap_in - h_evap_out)
```

```
" High and Low Concentration Mass Flow Rates "
m_dot_low_concentration + m_dot_Refrigerant = m_dot_high_concentration
m_dot_low_concentration*Conc_low + m_dot_Refrigerant*Conc_deph_out =
m_dot_high_concentration*Conc_high
" Pump Work "
CALL NH3H2O(123,T_abs_out + 273,Pressure_high,Conc_high: T_pump_out +
273,P_pump_out,x_pump_out,h_pump_out,s_pump_out,u_pump_out,v_pump_out,q_pump_out)
Q_pump_work1 = m_dot_high_concentration * (h_pump_out - h_abs_out)
Q_pump_work2 = m_dot_high_concentration * v_abs_out * (Pressure_high - Pressure_low) * 100
" Liquid Liquid Heat Exchanger "
CALL
LiqLiqHeatExch(Pressure_high,Pressure_low,Conc_high,Conc_low,eta_LLHX,h_pump_out,h_gen_liq_ou
tm_dot_high_concentration,m_dot_low_concentration,T_pump_out,T_gen_liq_out:Q_LLHX,T_gen_in,h_
gen_in,q_gen_in,T_abs_liq_in,h_abs_liq_in,q_abs_liq_in)
" Absorber Energy Requirements "
Q_absorber = m_dot_low_concentration*h_abs_liq_in + m_dot_Refrigerant*h_abs_vap_in -
m_dot_high_concentration*h_abs_out
" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high,Conc_deph_out,1: T_cond_in +
273,P_cond_in,x_cond_in,h_cond_in,s_cond_in,u_cond_in,v_cond_in,q_cond_in)
" Condenser Energy Requirements "
Q_condenser = m_dot_Refrigerant * (h_cond_in - h_cond_out)
" Generator Exit Conditions "
CALLGeneratorLiquidHeating(Conc_high,Pressure_high,Conc_low,m_dot_high_concentration:T_gen_va
p_out,h_gen_vap_out,x_gen_vap_out)
" Rectifier Analysis "
eta_Rectifier = (T_gen_vap_out - T_deph_in1)/(T_gen_vap_out - T_gen_in)
" Principal Operating Line "
CALL NH3H2O(128,T_deph_in1 + 273,Pressure_high,1: T_deph_in +
273,P_deph_in,x_deph_in,h_deph_in,s_deph_in,u_deph_in,v_deph_in,q_deph_in)
Slope = (h_deph_in - h_gen_in)/(x_deph_in - Conc_high)
h_deph_in = Slope*x_deph_in + constant

" Generator Energy Requirements "
Pole_Generator = Slope*Conc_low + constant
Q_generator = m_dot_low_concentration * (h_gen_liq_out - Pole_Generator)
" Dephlegmator Energy Requirements "
Pole_Dephlegmator = Slope*Conc_deph_out + constant
Q_dephlegmator = m_dot_Refrigerant * (Pole_Dephlegmator - h_cond_in)
" Coefficient of Performance "
COP = Refrigeration_Effect_Day/(Q_generator + Q_pump_work2)
" Energy Balance "
Energy_Balance_in = Refrigeration_Effect_Day + Q_generator + Q_pump_work1
Energy_Balance_out = Q_condenser + Q_absorber + Q_dephlegmator
```

APPENDIX A3 (EES Module for Alternative Design D1.2)

```
" Units deg C, bar, kJ, kg, m, sec, kW "
" sc  = solar collector  "
" ********************************  Input Data
************************************************** "
T_gen = 120
T_cond = 45
T_evap = -9
T_abs = 37
Conc_deph_out = 0.996
eta_VLHX = 0.5
Refrigeration_Effect_D&N = 5
Hours_Day = 10
" ******************************** SOLUTION
**************************************************"
Hours_Night = 24 - Hours_Day
" Required Refrigeration Effect in kJ"
Refrigeration_Effect_Day = Refrigeration_Effect_D&N * 3600 * 24
ICE_STORAGE = Hours_Day * 3600 * Refrigeration_Effect_D&N / Enthalpy_fusion(Water)
" Parameters Selection "
CALL NH3H2O(138,T_evap + 273,1,1: T_evap_out +
273,P_evap_out,x_evap_out,h_evap_out,s_evap_out,u_evap_out,v_evap_out,q_evap_out)
Pressure_low = P_evap_out
CALL NH3H2O(128,T_abs +273,Pressure_low,0: T_abs_out
+273,P_abs_out,x_abs_out,h_abs_out,s_abs_out,u_abs_out,v_abs_out,q_abs_out)
Conc_high = x_abs_out
CALL NH3H2O(138,T_cond + 273,Conc_deph_out,0: T_cond_out +
273,P_cond_out,x_cond_out,h_cond_out,s_cond_out,u_cond_out,v_cond_out,q_cond_out)
Pressure_high = P_cond_out
CALL NH3H2O(128,T_gen + 273,Pressure_high,0: T_gen_liq_out +
273,P_gen_liq_out,x_gen_liq_out,h_gen_liq_out,s_gen_liq_out,u_gen_liq_out,v_gen_liq_out,q_gen_liq_o
ut)Conc_low = x_gen_liq_out
CALL NH3H2O(238,Pressure_high,Conc_high,0: T_gen_start +
273,P_gen_start,x_gen_start,h_gen_start,s_gen_start,u_gen_start,v_gen_start,q_gen_start)
CALL NH3H2O(238,Pressure_low,Conc_low,0: T_abs_start +
273,P_abs_start,x_abs_start,h_abs_start,s_abs_start,u_abs_start,v_abs_start,q_abs_start)
" Vapor Liquid Heat Exchanger "
eta_VLHX = (T_vap_out_VLHX - T_evap_out)/(T_cond_out - T_evap_out)
CALL NH3H2O(123,T_vap_out_VLHX + 273,Pressure_low,1: T_abs_vap_in +
273,P_abs_vap_in,x_abs_vap_in,h_abs_vap_in,s_abs_vap_in,u_abs_vap_in,v_abs_vap_in,q_abs_vap_in)
h_liq_out_VLHX = h_cond_out - (h_abs_vap_in - h_evap_out)
CALL NH3H2O(234,Pressure_high,Conc_deph_out,h_liq_out_VLHX: T_before_expansion +
273,P_before_expansion,x_before_expansion,h_before_expansion,s_before_expansion,u_before_expansio
n,v_before_expansion,q_before_expansion)
" Liquid Ammonia Tank"
Refrigeration_Effect_Day = Mass_Refrigerant * (h_evap_out - h_after_expansion)
T_Liquid_Ammonia_Tank = T_cond
" Energy Exchange at VLHX "
E_VLHX1 = Mass_Refrigerant * (h_cond_out - h_liq_out_VLHX)
E_VLHX2 = Mass_Refrigerant * (h_abs_vap_in - h_evap_out)
" Intermittent System Mass "
CALL SC(Conc_high,Conc_low,Pressure_high,Conc_deph_out:Slope,Constant)
Mass_Refrigerant = Slope * Mass_Total + Constant
 " Ammonia Generation "
```

```
CALLCinCout(Conc_high,Pressure_high,Conc_low,Mass_Total,300:Total_gen_in,Total_gen_out,Total_gen_vap,Ammonia_gen_in,Ammonia_gen_out,
Ammonia_gen_vap,Water_gen_in,Water_gen_out,Water_gen_vap,T_gen_in,T_gen_out,T_gen_vap,h_gen_in,h_gen_out,h_gen_vap,x_gen_in,x_gen_out,x_gen_vap,E_generation)
" Dephlegmation Process "
CALLVCTinCout(T_gen_vap,Pressure_high,Conc_deph_out,Total_gen_vap,300:Total_deph_in,Total_deph_out,Total_deph_con,Ammonia_deph_in,Ammonia_deph_out,
Ammonia_deph_con,Water_deph_in,Water_deph_out,Water_deph_con,T_deph_in,T_deph_out,T_deph_con,h_deph_in,h_deph_out,h_deph_con,x_deph_in,x_deph_out,x_deph_con,E_dephlegmation)
" Reflux Tank "
T_Reflux_Tank = T_deph_con
Mass_Reflux_Tank = Total_deph_con
x_Reflux_Tank = x_deph_con
" Pressurization Process"
T_gen_initial = T_abs
CALL NH3H2O(138,T_gen_initial + 273,Conc_high,0: T_bp +273,P_bp,x_bp,h_bp,s_bp,u_bp,v_bp,q_bp)
"Before Pressurization"
CALL NH3H2O(238,Pressure_high,Conc_high,0: T_ap +273,P_ap,x_ap,h_ap,s_ap,u_ap,v_ap,q_ap) "After Pressurization"
E_pressurization = (h_ap - h_bp)*Mass_Total
" De-Pressurization Process"
T_abs_initial = T_gen
CALL NH3H2O(138,T_abs_initial + 273,Conc_low,0: T_bdp
+273,P_bdp,x_bdp,h_bdp,s_bdp,u_bdp,v_bdp,q_bdp) "Before De-Pressurization"
CALL NH3H2O(238,Pressure_low,Conc_low,0: T_adp
+273,P_adp,x_adp,h_adp,s_adp,u_adp,v_adp,q_adp) "After De-Pressurization"
E_Depressurization = (h_bdp - h_adp)*Total_gen_out
" Condensation Process "
CALL NH3H2O(123,T_deph_out +273,Pressure_high,Conc_deph_out: T_bc
+273,P_bc,x_bc,h_bc,s_bc,u_bc,v_bc,q_bc)          "Before Condensation"
CALL NH3H2O(238,Pressure_high,Conc_deph_out,0: T_ac +273,P_ac,x_ac,h_ac,s_ac,u_ac,v_ac,q_ac)
"After Condensation"
E_condensation = (h_bc - h_ac)*Mass_Refrigerant
" Absorption Process "
CALL NH3H2O(123,T_adp + 273,Pressure_low,Conc_low: T_abs_start_ACTUAL
+273,P_abs_start_ACTUAL,x_abs_start_ACTUAL,h_abs_start_ACTUAL,s_abs_start_ACTUAL,u_abs_start_ACTUAL,v_abs_start_ACTUAL,q_abs_start_ACTUAL)  "ACTUAL Absorption Start"
CALL NH3H2O(238,Pressure_low,Conc_high,0: T_abs_end_ACTUAL
+273,P_abs_end_ACTUAL,x_abs_end_ACTUAL,h_abs_end_ACTUAL,s_abs_end_ACTUAL,u_abs_end_ACTUAL,v_abs_end_ACTUAL,q_abs_end_ACTUAL)        "ACTUAL Absorption End"
Total_gen_out *h_abs_start_ACTUAL + Mass_Refrigerant * h_abs_vap_in = E_absorption +
(Mass_Refrigerant + Total_gen_out)*h_abs_end_ACTUAL
" Energy Balance "
Energy_Balance_in = E_generation + E_pressurization + Refrigeration_Effect_Day
Energy_Balance_out = E_absorption + E_Depressurization + E_condensation + E_dephlegmation
" Coefficient of Performance "
COP = Refrigeration_Effect_Day/(E_generation + E_pressurization)
" Heat Duties "
Q_generator = (E_generation + E_pressurization)/(Hours_Day * 3600)
Q_dephlegmator = (E_dephlegmation)/(Hours_Day * 3600)
Q_condenser = (E_condensation)/(Hours_Day * 3600)
Q_absorber = (E_absorption + E_Depressurization)/(Hours_Night * 3600)
Q_VLHX = E_VLHX1/((Hours_Night * 3600) *(E_absorption/(E_absorption + E_Depressurization)))
Q_evaporator = Refrigeration_Effect_Day/((Hours_Night * 3600) *(E_absorption/(E_absorption +
E_Depressurization)))
```

APPENDIX A4 (EES Module for Alternative Design D2.1)

```
T_gen = 120
T_cond = 45
T_evap = -9
T_abs_DAY = 45
T_abs_NIGHT = 37
Conc_deph_out = 0.996
eta_LLHX = 0.8
eta_VLHX = 0.5
eta_Rectifier = 0.75
Refrigeration_Effect_D&N = 5
Hours_Day = 10
Hours_Night = 24 - Hours_Day
" Required Refrigeration Effect"
Refrigeration_Effect_Day = Refrigeration_Effect_D&N*24/Hours_Day
{ICE_STORAGE = (24 - Hours_Day) * 3600 * Refrigeration_Effect_D&N / Enthalpy_fusion(Water)}
" Parameters Selection "
CALL NH3H2O(138,T_evap + 273,1,1: T_evap_out +
273,P_evap_out,x_evap_out,h_evap_out,s_evap_out,u_evap_out,v_evap_out,q_evap_out)
Pressure_low = P_evap_out
CALL NH3H2O(128,T_abs_DAY +273,Pressure_low,0: T_abs_out_DAY
+273,P_abs_out_DAY,x_abs_out_DAY,h_abs_out_DAY,s_abs_out_DAY,u_abs_out_DAY,v_abs_out_D
AY,q_abs_out_DAY)
Conc_high_DAY = x_abs_out_DAY
CALL NH3H2O(128,T_abs_NIGHT +273,Pressure_low,0: T_abs_out_NIGHT
+273,P_abs_out_NIGHT,x_abs_out_NIGHT,h_abs_out_NIGHT,s_abs_out_NIGHT,u_abs_out_NIGHT,v_
abs_out_NIGHT,q_abs_out_NIGHT)
Conc_high_NIGHT = x_abs_out_NIGHT
CALL NH3H2O(138,T_cond + 273,Conc_deph_out,0: T_cond_out +
273,P_cond_out,x_cond_out,h_cond_out,s_cond_out,u_cond_out,v_cond_out,q_cond_out)
Pressure_high = P_cond_out
CALL NH3H2O(128,T_gen + 273,Pressure_high,0: T_gen_liq_out +
273,P_gen_liq_out,x_gen_liq_out,h_gen_liq_out,s_gen_liq_out,u_gen_liq_out,v_gen_liq_out,q_gen_liq_o
ut)
Conc_low = x_gen_liq_out
CALL NH3H2O(238,Pressure_high,Conc_high_DAY,0: T_gen_start_DAY +
273,P_gen_start_DAY,x_gen_start_DAY,h_gen_start_DAY,s_gen_start_DAY,u_gen_start_DAY,v_gen_s
tart_DAY,q_gen_start_DAY)
CALL NH3H2O(238,Pressure_high,Conc_high_NIGHT,0: T_gen_start_NIGHT +
273,P_gen_start_NIGHT,x_gen_start_NIGHT,h_gen_start_NIGHT,s_gen_start_NIGHT,u_gen_start_NIG
HT,v_gen_start_NIGHT,q_gen_start_NIGHT)
CALL NH3H2O(238,Pressure_low,Conc_low,0: T_abs_start +
273,P_abs_start,x_abs_start,h_abs_start,s_abs_start,u_abs_start,v_abs_start,q_abs_start)
" Vapor Liquid Heat Exchanger "
eta_VLHX = (T_vap_out_VLHX - T_evap_out)/(T_cond_out - T_evap_out)
CALL NH3H2O(123,T_vap_out_VLHX + 273,Pressure_low,1: T_abs_vap_in +
273,P_abs_vap_in,x_abs_vap_in,h_abs_vap_in,s_abs_vap_in,u_abs_vap_in,v_abs_vap_in,q_abs_vap_in)
h_liq_out_VLHX = h_cond_out - (h_abs_vap_in - h_evap_out)
CALL NH3H2O(234,Pressure_high,Conc_deph_out,h_liq_out_VLHX: T_before_expansion +
273,P_before_expansion,x_before_expansion,h_before_expansion,s_before_expansion,u_before_expansio
n,v_before_expansion,q_before_expansion)
" Ammonia Expansion Process "
```

```
CALL NH3H2O(234,Pressure_low,Conc_deph_out,h_before_expansion: T_after_expansion +
273,P_after_expansion,x_after_expansion,h_after_expansion,s_after_expansion,u_after_expansion,v_after
_expansion,q_after_expansion)
" Refrigerant Flow Rate"
Refrigeration_Effect_Day = m_dot_Refrigerant_HPZ * (h_evap_out - h_after_expansion)
Refrigeration_Effect_D&N = m_dot_Refrigerant_LPZ * (h_evap_out - h_after_expansion)
" Energy Exchange at VLHX "
Q_VLHX1 = m_dot_Refrigerant_LPZ * (h_cond_out - h_liq_out_VLHX)
Q_VLHX2 = m_dot_Refrigerant_LPZ * (h_abs_vap_in - h_evap_out)
" High and Low Concentration Mass Flow Rates "
m_dot_low_conc_abs_DAY + m_dot_Refrigerant_LPZ = m_dot_high_conc_abs_DAY
m_dot_low_conc_abs_DAY*Conc_low + m_dot_Refrigerant_LPZ*Conc_deph_out =
m_dot_high_conc_abs_DAY*Conc_high_DAY
m_dot_low_conc_abs_NIGHT + m_dot_Refrigerant_LPZ = m_dot_high_conc_abs_NIGHT
m_dot_low_conc_abs_NIGHT*Conc_low + m_dot_Refrigerant_LPZ*Conc_deph_out =
m_dot_high_conc_abs_NIGHT*Conc_high_NIGHT
" Storages Volume "
Storage_Liquid_Ammonia = ( m_dot_Refrigerant_HPZ - m_dot_Refrigerant_LPZ ) * 3600 * Hours_Day
Storage_Weak_Solution = m_dot_low_conc_abs_NIGHT * 3600 * Hours_Night
Storage_Strong_Solution = m_dot_high_conc_abs_NIGHT * 3600 * Hours_Night
" Pump Work "
m_dot_high_conc_StrongSol = Storage_Strong_Solution/(Hours_Day * 3600)    "Extraction From Strong
Solution Tank at the Day time"
m_dot_high_concentration = m_dot_high_conc_abs_DAY + m_dot_high_conc_StrongSol
m_dot_low_conc_WeakSol = Storage_Weak_Solution/(Hours_Day * 3600)       "Supply to Weak Solution
Tank at the Day time "
m_dot_low_concentration = m_dot_low_conc_abs_DAY + m_dot_low_conc_WeakSol
m_dot_high_concentration * Conc_pump_in = m_dot_high_conc_abs_DAY * Conc_high_DAY +
m_dot_high_conc_StrongSol * Conc_high_NIGHT
m_dot_high_concentration * h_pump_in1 = m_dot_high_conc_abs_DAY * h_abs_out_DAY +
m_dot_high_conc_StrongSol * h_abs_out_NIGHT
CALL NH3H2O(234,Pressure_low,Conc_pump_in,h_pump_in1: T_pump_in +
273,P_pump_in,x_pump_in,h_pump_in,s_pump_in,u_pump_in,v_pump_in,q_pump_in)
CALL NH3H2O(123,T_pump_in + 273,Pressure_high,Conc_pump_in: T_pump_out +
273,P_pump_out,x_pump_out,h_pump_out,s_pump_out,u_pump_out,v_pump_out,q_pump_out)
Q_pump_work1 = m_dot_high_concentration * (h_pump_out - h_pump_in)
Q_pump_work2 = m_dot_high_concentration * v_pump_in * (Pressure_high - Pressure_low) * 100
CALL
LiqLiqHeatExch(Pressure_high,Pressure_low,Conc_pump_in,Conc_low,eta_LLHX,h_pump_out,h_gen_li
q_out
,m_dot_high_concentration,m_dot_low_concentration,T_pump_out,T_gen_liq_out:Q_LLHX,T_gen_in,h_
gen_in,q_gen_in,T_abs_liq_in,h_abs_liq_in,q_abs_liq_in)
" Absorber Energy Requirements "
Q_absorber_DAY = m_dot_low_conc_abs_DAY*h_abs_liq_in + m_dot_Refrigerant_LPZ*h_abs_vap_in -
m_dot_high_conc_abs_DAY*h_abs_out_DAY
Q_absorber_NIGHT = m_dot_low_conc_abs_NIGHT*h_abs_liq_in +
m_dot_Refrigerant_LPZ*h_abs_vap_in - m_dot_high_conc_abs_NIGHT*h_abs_out_NIGHT
" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high,Conc_deph_out,1: T_cond_in +
273,P_cond_in,x_cond_in,h_cond_in,s_cond_in,u_cond_in,v_cond_in,q_cond_in)
" Condenser Energy Requirements "
Q_condenser = m_dot_Refrigerant_HPZ * (h_cond_in - h_cond_out)
" Generator Exit Conditions "
CALL
GeneratorLiquidHeating(Conc_pump_in,Pressure_high,Conc_low,m_dot_high_concentration:T_gen_vap_
out,h_gen_vap_out,x_gen_vap_out)
```

```
CALL NH3H2O(238,Pressure_high,Conc_pump_in,0: T_gen_start_ACTUAL +
273,P_gen_start_ACTUAL,x_gen_start_ACTUAL,h_gen_start_ACTUAL,s_gen_start_ACTUAL,u_gen_st
art_ACTUAL,v_gen_start_ACTUAL,q_gen_start_ACTUAL)
" Rectifier Analysis "
eta_Rectifier = (T_gen_vap_out - T_deph_in1)/(T_gen_vap_out - T_gen_in)
" Principal Operating Line "
CALL NH3H2O(128,T_deph_in1 + 273,Pressure_high,1: T_deph_in +
273,P_deph_in,x_deph_in,h_deph_in,s_deph_in,u_deph_in,v_deph_in,q_deph_in)
Slope = (h_deph_in - h_gen_in)/(x_deph_in - Conc_pump_in)
h_deph_in = Slope*x_deph_in + constant
" Generator Energy Requirements "
Pole_Generator = Slope*Conc_low + constant
Q_generator = m_dot_low_concentration * (h_gen_liq_out - Pole_Generator)
" Dephlegmator Energy Requirements "
Pole_Dephlegmator = Slope*Conc_deph_out + constant
Q_dephlegmator = m_dot_Refrigerant_HPZ * (Pole_Dephlegmator - h_cond_in)
" Coefficient of Performance "
COP = Refrigeration_Effect_Day/(Q_generator + Q_pump_work2)
" Energy Balance "
Energy_Balance_in = (Refrigeration_Effect_Day + Q_generator + Q_pump_work1)*Hours_Day
Energy_Balance_out = Q_condenser*Hours_Day + Q_absorber_DAY*Hours_Day +
Q_absorber_NIGHT*Hours_Night + Q_dephlegmator*Hours_Day
```

APPENDIX A5 (EES Module for Alternative Design D2.2)

```
T_gen_2S = 80
T_cond_2S = 22
T_evap_2S = -9
T_abs_DAY_2S = 22
T_abs_NIGHT_2S = 22
Conc_deph_out_2S = 0.996
eta_LLHX_2S = 0.8
eta_VLHX_2S = 0.5
eta_Rectifier_2S = 0.75
Refrigeration_Effect_D&N_2S = 5
Hours_Day_2S = 10
Hours_Night_2S = 24 - Hours_Day_2S
" Required Refrigeration Effect"
Refrigeration_Effect_Day_2S = Refrigeration_Effect_D&N_2S*24/Hours_Day_2S
" Parameters Selection "
CALL NH3H2O(138,T_evap_2S + 273,1,1: T_evap_out_2S +
273,P_evap_out_2S,x_evap_out_2S,h_evap_out_2S,s_evap_out_2S,u_evap_out_2S,v_evap_out_2S,q_eva
p_out_2S)
Pressure_low_2S = P_evap_out_2S
CALL NH3H2O(128,T_abs_DAY_2S +273,Pressure_low_2S,0: T_abs_out_DAY_2S
+273,P_abs_out_DAY_2S,x_abs_out_DAY_2S,h_abs_out_DAY_2S,s_abs_out_DAY_2S,u_abs_out_DA
Y_2S,v_abs_out_DAY_2S,q_abs_out_DAY_2S)
Conc_high_DAY_2S = x_abs_out_DAY_2S
CALL NH3H2O(128,T_abs_NIGHT_2S +273,Pressure_low_2S,0: T_abs_out_NIGHT_2S
+273,P_abs_out_NIGHT_2S,x_abs_out_NIGHT_2S,h_abs_out_NIGHT_2S,s_abs_out_NIGHT_2S,u_abs
_out_NIGHT_2S,v_abs_out_NIGHT_2S,q_abs_out_NIGHT_2S)
Conc_high_NIGHT_2S = x_abs_out_NIGHT_2S
CALL NH3H2O(138,T_cond_2S + 273,Conc_deph_out_2S,0: T_cond_out_2S +
273,P_cond_out_2S,x_cond_out_2S,h_cond_out_2S,s_cond_out_2S,u_cond_out_2S,v_cond_out_2S,q_co
nd_out_2S)
Pressure_high_2S = P_cond_out_2S
CALL NH3H2O(128,T_gen_2S + 273,Pressure_high_2S,0: T_gen_liq_out_2S +
273,P_gen_liq_out_2S,x_gen_liq_out_2S,h_gen_liq_out_2S,s_gen_liq_out_2S,u_gen_liq_out_2S,v_gen_li
q_out_2S,q_gen_liq_out_2S)
Conc_low_2S = x_gen_liq_out_2S
CALL NH3H2O(238,Pressure_high_2S,Conc_high_DAY_2S,0: T_gen_start_DAY_2S +
273,P_gen_start_DAY_2S,x_gen_start_DAY_2S,h_gen_start_DAY_2S,s_gen_start_DAY_2S,u_gen_start
_DAY_2S,v_gen_start_DAY_2S,q_gen_start_DAY_2S)
CALL NH3H2O(238,Pressure_high_2S,Conc_high_NIGHT_2S,0: T_gen_start_NIGHT_2S +
273,P_gen_start_NIGHT_2S,x_gen_start_NIGHT_2S,h_gen_start_NIGHT_2S,s_gen_start_NIGHT_2S,u_
gen_start_NIGHT_2S,v_gen_start_NIGHT_2S,q_gen_start_NIGHT_2S)
" In order to determine the actual temperature at which generation starts, see below Generator Exit
Conditions "
CALL NH3H2O(238,Pressure_low_2S,Conc_low_2S,0: T_abs_start_2S +
273,P_abs_start_2S,x_abs_start_2S,h_abs_start_2S,s_abs_start_2S,u_abs_start_2S,v_abs_start_2S,q_abs_s
tart_2S)
" Vapor Liquid Heat Exchanger "
eta_VLHX_2S = (T_vap_out_VLHX_2S - T_evap_out_2S)/(T_cond_out_2S - T_evap_out_2S)
CALL NH3H2O(123,T_vap_out_VLHX_2S + 273,Pressure_low_2S,1: T_abs_vap_in_2S +
273,P_abs_vap_in_2S,x_abs_vap_in_2S,h_abs_vap_in_2S,s_abs_vap_in_2S,u_abs_vap_in_2S,v_abs_vap
_in_2S,q_abs_vap_in_2S)
h_liq_out_VLHX_2S = h_cond_out_2S - (h_abs_vap_in_2S - h_evap_out_2S)
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```
CALL NH3H2O(234,Pressure_high_2S,Conc_deph_out_2S,h_liq_out_VLHX_2S:
T_before_expansion_2S +
273,P_before_expansion_2S,x_before_expansion_2S,h_before_expansion_2S,s_before_expansion_2S,u_b
efore_expansion_2S,v_before_expansion_2S,q_before_expansion_2S)
" Ammonia Expansion Process "
CALL NH3H2O(234,Pressure_low_2S,Conc_deph_out_2S,h_before_expansion_2S:
T_after_expansion_2S +
273,P_after_expansion_2S,x_after_expansion_2S,h_after_expansion_2S,s_after_expansion_2S,u_after_ex
pansion_2S,v_after_expansion_2S,q_after_expansion_2S)
" Refrigerant Flow Rate"
Refrigeration_Effect_Day_2S = m_dot_Refrigerant_HPZ_2S * (h_evap_out_2S - h_after_expansion_2S)
Refrigeration_Effect_D&N_2S = m_dot_Refrigerant_LPZ_2S * (h_evap_out_2S - h_after_expansion_2S)
" Energy Exchange at VLHX "
Q_VLHX1_2S = m_dot_Refrigerant_LPZ_2S * (h_cond_out_2S - h_liq_out_VLHX_2S)
Q_VLHX2_2S = m_dot_Refrigerant_LPZ_2S * (h_abs_vap_in_2S - h_evap_out_2S)
" High and Low Concentration Mass Flow Rates "
m_dot_low_conc_abs_DAY_2S + m_dot_Refrigerant_LPZ_2S = m_dot_high_conc_abs_DAY_2S
m_dot_low_conc_abs_DAY_2S*Conc_low_2S + m_dot_Refrigerant_LPZ_2S*Conc_deph_out_2S =
m_dot_high_conc_abs_DAY_2S*Conc_high_DAY_2S
m_dot_low_conc_abs_NIGHT_2S + m_dot_Refrigerant_LPZ_2S = m_dot_high_conc_abs_NIGHT_2S
m_dot_low_conc_abs_NIGHT_2S*Conc_low_2S + m_dot_Refrigerant_LPZ_2S*Conc_deph_out_2S =
m_dot_high_conc_abs_NIGHT_2S*Conc_high_NIGHT_2S
" Storages Volume "
Storage_Liquid_Ammonia_2S = ( m_dot_Refrigerant_HPZ_2S - m_dot_Refrigerant_LPZ_2S ) * 3600 *
Hours_Day_2S
Storage_Weak_Solution_2S = m_dot_low_conc_abs_NIGHT_2S * 3600 * Hours_Night_2S
Storage_Strong_Solution_2S = m_dot_high_conc_abs_NIGHT_2S * 3600 * Hours_Night_2S
" Pump Work "
m_dot_high_conc_StrongSol_2S = Storage_Strong_Solution_2S/(Hours_Day_2S * 3600)    "Extraction
From Strong Solution Tank at the Day time"
m_dot_high_concentration_2S = m_dot_high_conc_abs_DAY_2S + m_dot_high_conc_StrongSol_2S
m_dot_low_conc_WeakSol_2S = Storage_Weak_Solution_2S/(Hours_Day_2S * 3600)       "Supply to
Weak Solution Tank at the Day time "
m_dot_low_concentration_2S = m_dot_low_conc_abs_DAY_2S + m_dot_low_conc_WeakSol_2S
m_dot_high_concentration_2S * Conc_pump_in_2S = m_dot_high_conc_abs_DAY_2S *
Conc_high_DAY_2S + m_dot_high_conc_StrongSol_2S * Conc_high_NIGHT_2S
m_dot_high_concentration_2S * h_pump_in1_2S = m_dot_high_conc_abs_DAY_2S *
h_abs_out_DAY_2S + m_dot_high_conc_StrongSol_2S * h_abs_out_NIGHT_2S
CALL NH3H2O(234,Pressure_low_2S,Conc_pump_in_2S,h_pump_in1_2S: T_pump_in_2S +
273,P_pump_in_2S,x_pump_in_2S,h_pump_in_2S,s_pump_in_2S,u_pump_in_2S,v_pump_in_2S,q_pump
_in_2S)
CALL NH3H2O(123,T_pump_in_2S + 273,Pressure_high_2S,Conc_pump_in_2S: T_pump_out_2S +
273,P_pump_out_2S,x_pump_out_2S,h_pump_out_2S,s_pump_out_2S,u_pump_out_2S,v_pump_out_2S,
q_pump_out_2S)
Q_pump_work1_2S = m_dot_high_concentration_2S * (h_pump_out_2S - h_pump_in_2S)
Q_pump_work2_2S = m_dot_high_concentration_2S * v_pump_in_2S * (Pressure_high_2S -
Pressure_low_2S) * 100
CALL
LiqLiqHeatExch(Pressure_high_2S,Pressure_low_2S,Conc_pump_in_2S,Conc_low_2S,eta_LLHX_2S,h_
pump_out_2S,h_gen_liq_out_2S
,m_dot_high_concentration_2S,m_dot_low_concentration_2S,T_pump_out_2S,T_gen_liq_out_2S:Q_LLH
X_2S,T_gen_in_2S,h_gen_in_2S,q_gen_in_2S,T_abs_liq_in_2S,h_abs_liq_in_2S,q_abs_liq_in_2S)
" Absorber Energy Requirements "
Q_absorber_DAY_2S = m_dot_low_conc_abs_DAY_2S*h_abs_liq_in_2S +
m_dot_Refrigerant_LPZ_2S*h_abs_vap_in_2S - m_dot_high_conc_abs_DAY_2S*h_abs_out_DAY_2S
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Q_absorber_NIGHT_2S = m_dot_low_conc_abs_NIGHT_2S*h_abs_liq_in_2S +
m_dot_Refrigerant_LPZ_2S*h_abs_vap_in_2S -
m_dot_high_conc_abs_NIGHT_2S*h_abs_out_NIGHT_2S
" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high_2S,Conc_deph_out_2S,1: T_cond_in_2S +
273,P_cond_in_2S,x_cond_in_2S,h_cond_in_2S,s_cond_in_2S,u_cond_in_2S,v_cond_in_2S,q_cond_in_2
S)
" Condenser Energy Requirements "
Q_condenser_2S = m_dot_Refrigerant_HPZ_2S * (h_cond_in_2S - h_cond_out_2S)
" Generator Exit Conditions "
CALL
GeneratorLiquidHeating(Conc_pump_in_2S,Pressure_high_2S,Conc_low_2S,m_dot_high_concentration_
2S:T_gen_vap_out_2S,h_gen_vap_out_2S,x_gen_vap_out_2S)
CALL NH3H2O(238,Pressure_high_2S,Conc_pump_in_2S,0: T_gen_start_ACTUAL_2S +
273,P_gen_start_ACTUAL_2S,x_gen_start_ACTUAL_2S,h_gen_start_ACTUAL_2S,s_gen_start_ACTU
AL_2S,u_gen_start_ACTUAL_2S,v_gen_start_ACTUAL_2S,q_gen_start_ACTUAL_2S)
" Rectifier Analysis "
eta_Rectifier_2S = (T_gen_vap_out_2S - T_deph_in1_2S)/(T_gen_vap_out_2S - T_gen_in_2S)
" Prinipal Operating Line "
CALL NH3H2O(128,T_deph_in1_2S + 273,Pressure_high_2S,1: T_deph_in_2S +
273,P_deph_in_2S,x_deph_in_2S,h_deph_in_2S,s_deph_in_2S,u_deph_in_2S,v_deph_in_2S,q_deph_in_2
S)
Slope_2S = (h_deph_in_2S - h_gen_in_2S)/(x_deph_in_2S - Conc_pump_in_2S)
h_deph_in_2S = Slope_2S*x_deph_in_2S + constant_2S
" Generator Energy Requirements "
Pole_Generator_2S = Slope_2S*Conc_low_2S + constant_2S
Q_generator_2S = m_dot_low_concentration_2S * (h_gen_liq_out_2S - Pole_Generator_2S)
" Dephlegmator Energy Requirements "
Pole_Dephlegmator_2S = Slope_2S*Conc_deph_out_2S + constant_2S
Q_dephlegmator_2S = m_dot_Refrigerant_HPZ_2S * (Pole_Dephlegmator_2S - h_cond_in_2S)
" Coefficient of Performance "
COP_2S = Refrigeration_Effect_Day_2S/(Q_generator_2S + Q_pump_work2_2S)
" Energy Balance "
Energy_Balance_in_2S = (Refrigeration_Effect_Day_2S + Q_generator_2S +
Q_pump_work1_2S)*Hours_Day_2S
Energy_Balance_out_2S = Q_condenser_2S*Hours_Day_2S + Q_absorber_DAY_2S*Hours_Day_2S +
Q_absorber_NIGHT_2S*Hours_Night_2S + Q_dephlegmator_2S*Hours_Day_2S
"! ======================================== SECOND STAGE END
========================================="
"! ======================================== FIRST STAGE START
========================================="
T_gen_1S = T_gen_2S
T_cond_1S = 45
T_evap_1S = T_abs_NIGHT_2S - 5
T_abs_DAY_1S = 45
T_abs_NIGHT_1S = 37
Conc_deph_out_1S = 0.996
eta_LLHX_1S = 0.8
eta_VLHX_1S = 0.5
eta_Rectifier_1S = 0.75
Refrigeration_Effect_D&N_1S = (Q_condenser_2S*Hours_Day_2S +
Q_absorber_DAY_2S*Hours_Day_2S + Q_absorber_NIGHT_2S*Hours_Night_2S +
Q_dephlegmator_2S*Hours_Day_2S)/24
Hours_Day_1S = 10
```

```
" ****************************** SOLUTION
************************************************"
Hours_Night_1S = 24 - Hours_Day_1S
" Required Refrigeration Effect"
Refrigeration_Effect_Day_1S = Refrigeration_Effect_D&N_1S*24/Hours_Day_1S
" Parameters Selection "
CALL NH3H2O(138,T_evap_1S + 273,1,1: T_evap_out_1S +
273,P_evap_out_1S,x_evap_out_1S,h_evap_out_1S,s_evap_out_1S,u_evap_out_1S,v_evap_out_1S,q_eva
p_out_1S)
Pressure_low_1S = P_evap_out_1S
CALL NH3H2O(128,T_abs_DAY_1S +273,Pressure_low_1S,0: T_abs_out_DAY_1S
+273,P_abs_out_DAY_1S,x_abs_out_DAY_1S,h_abs_out_DAY_1S,s_abs_out_DAY_1S,u_abs_out_DA
Y_1S,v_abs_out_DAY_1S,q_abs_out_DAY_1S)
Conc_high_DAY_1S = x_abs_out_DAY_1S
CALL NH3H2O(128,T_abs_NIGHT_1S +273,Pressure_low_1S,0: T_abs_out_NIGHT_1S
+273,P_abs_out_NIGHT_1S,x_abs_out_NIGHT_1S,h_abs_out_NIGHT_1S,s_abs_out_NIGHT_1S,u_abs
_out_NIGHT_1S,v_abs_out_NIGHT_1S,q_abs_out_NIGHT_1S)
Conc_high_NIGHT_1S = x_abs_out_NIGHT_1S
CALL NH3H2O(138,T_cond_1S + 273,Conc_deph_out_1S,0: T_cond_out_1S +
273,P_cond_out_1S,x_cond_out_1S,h_cond_out_1S,s_cond_out_1S,u_cond_out_1S,v_cond_out_1S,q_co
nd_out_1S)
Pressure_high_1S = P_cond_out_1S
CALL NH3H2O(128,T_gen_1S + 273,Pressure_high_1S,0: T_gen_liq_out_1S +
273,P_gen_liq_out_1S,x_gen_liq_out_1S,h_gen_liq_out_1S,s_gen_liq_out_1S,u_gen_liq_out_1S,v_gen_li
q_out_1S,q_gen_liq_out_1S)
Conc_low_1S = x_gen_liq_out_1S
CALL NH3H2O(238,Pressure_high_1S,Conc_high_DAY_1S,0: T_gen_start_DAY_1S +
273,P_gen_start_DAY_1S,x_gen_start_DAY_1S,h_gen_start_DAY_1S,s_gen_start_DAY_1S,u_gen_start
_DAY_1S,v_gen_start_DAY_1S,q_gen_start_DAY_1S)
CALL NH3H2O(238,Pressure_high_1S,Conc_high_NIGHT_1S,0: T_gen_start_NIGHT_1S +
273,P_gen_start_NIGHT_1S,x_gen_start_NIGHT_1S,h_gen_start_NIGHT_1S,s_gen_start_NIGHT_1S,u_
gen_start_NIGHT_1S,v_gen_start_NIGHT_1S,q_gen_start_NIGHT_1S)
" In order to determine the actual temperature at which generation starts, see below Generator Exit
Conditions "
CALL NH3H2O(238,Pressure_low_1S,Conc_low_1S,0: T_abs_start_1S +
273,P_abs_start_1S,x_abs_start_1S,h_abs_start_1S,s_abs_start_1S,u_abs_start_1S,v_abs_start_1S,q_abs_s
tart_1S)
" Vapor Liquid Heat Exchanger "
eta_VLHX_1S = (T_vap_out_VLHX_1S - T_evap_out_1S)/(T_cond_out_1S - T_evap_out_1S)
CALL NH3H2O(123,T_vap_out_VLHX_1S + 273,Pressure_low_1S,1: T_abs_vap_in_1S +
273,P_abs_vap_in_1S,x_abs_vap_in_1S,h_abs_vap_in_1S,s_abs_vap_in_1S,u_abs_vap_in_1S,v_abs_vap
_in_1S,q_abs_vap_in_1S)
h_liq_out_VLHX_1S = h_cond_out_1S - (h_abs_vap_in_1S - h_evap_out_1S)
CALL NH3H2O(234,Pressure_high_1S,Conc_deph_out_1S,h_liq_out_VLHX_1S:
T_before_expansion_1S +
273,P_before_expansion_1S,x_before_expansion_1S,h_before_expansion_1S,s_before_expansion_1S,u_b
efore_expansion_1S,v_before_expansion_1S,q_before_expansion_1S)
" Ammonia Expansion Process "
CALL NH3H2O(234,Pressure_low_1S,Conc_deph_out_1S,h_before_expansion_1S:
T_after_expansion_1S +
273,P_after_expansion_1S,x_after_expansion_1S,h_after_expansion_1S,s_after_expansion_1S,u_after_ex
pansion_1S,v_after_expansion_1S,q_after_expansion_1S)
" Refrigerant Flow Rate"
Refrigeration_Effect_Day_1S = m_dot_Refrigerant_HPZ_1S * (h_evap_out_1S - h_after_expansion_1S)
Ref_Effect_DayONLY_1S = Q_condenser_2S + Q_absorber_DAY_2S + Q_dephlegmator_2S
Ref_Effect_NightONLY_1S = Q_absorber_NIGHT_2S
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Ref_Effect_DayONLY_1S = m_dot_Refrigerant_LPZ_Day_1S * (h_evap_out_1S -
h_after_expansion_1S)
Ref_Effect_NightONLY_1S = m_dot_Refrigerant_LPZ_Night_1S * (h_evap_out_1S -
h_after_expansion_1S)
" Energy Exchange at VLHX "
Q_VLHX1_Day_1S = m_dot_Refrigerant_LPZ_Day_1S * (h_cond_out_1S - h_liq_out_VLHX_1S)
Q_VLHX2_Day_1S = m_dot_Refrigerant_LPZ_Day_1S * (h_abs_vap_in_1S - h_evap_out_1S)
Q_VLHX1_Night_1S = m_dot_Refrigerant_LPZ_Night_1S * (h_cond_out_1S - h_liq_out_VLHX_1S)
Q_VLHX2_Night_1S = m_dot_Refrigerant_LPZ_Night_1S * (h_abs_vap_in_1S - h_evap_out_1S)
" High and Low Concentration Mass Flow Rates "
m_dot_low_conc_abs_DAY_1S + m_dot_Refrigerant_LPZ_Day_1S = m_dot_high_conc_abs_DAY_1S
m_dot_low_conc_abs_DAY_1S*Conc_low_1S + m_dot_Refrigerant_LPZ_Day_1S*Conc_deph_out_1S =
m_dot_high_conc_abs_DAY_1S*Conc_high_DAY_1S
m_dot_low_conc_abs_NIGHT_1S + m_dot_Refrigerant_LPZ_Night_1S =
m_dot_high_conc_abs_NIGHT_1S
m_dot_low_conc_abs_NIGHT_1S*Conc_low_1S +
m_dot_Refrigerant_LPZ_Night_1S*Conc_deph_out_1S =
m_dot_high_conc_abs_NIGHT_1S*Conc_high_NIGHT_1S
" Storages Volume "
Storage_Liquid_Ammonia_1S = ( m_dot_Refrigerant_HPZ_1S - m_dot_Refrigerant_LPZ_Day_1S ) *
3600 * Hours_Day_1S
Storage_Weak_Solution_1S = m_dot_low_conc_abs_NIGHT_1S * 3600 * Hours_Night_1S
Storage_Strong_Solution_1S = m_dot_high_conc_abs_NIGHT_1S * 3600 * Hours_Night_1S
" Pump Work "
m_dot_high_conc_StrongSol_1S = Storage_Strong_Solution_1S/(Hours_Day_1S * 3600)    "Extraction
From Strong Solution Tank at the Day time"
m_dot_high_concentration_1S = m_dot_high_conc_abs_DAY_1S + m_dot_high_conc_StrongSol_1S
m_dot_low_conc_WeakSol_1S = Storage_Weak_Solution_1S/(Hours_Day_1S * 3600)       "Supply to
Weak Solution Tank at the Day time "
m_dot_low_concentration_1S = m_dot_low_conc_abs_DAY_1S + m_dot_low_conc_WeakSol_1S
m_dot_high_concentration_1S * Conc_pump_in_1S = m_dot_high_conc_abs_DAY_1S *
Conc_high_DAY_1S + m_dot_high_conc_StrongSol_1S * Conc_high_NIGHT_1S
m_dot_high_concentration_1S * h_pump_in1_1S = m_dot_high_conc_abs_DAY_1S *
h_abs_out_DAY_1S + m_dot_high_conc_StrongSol_1S * h_abs_out_NIGHT_1S
CALL NH3H2O(234,Pressure_low_1S,Conc_pump_in_1S,h_pump_in1_1S: T_pump_in_1S +
273,P_pump_in_1S,x_pump_in_1S,h_pump_in_1S,s_pump_in_1S,u_pump_in_1S,v_pump_in_1S,q_pump
_in_1S)
CALL NH3H2O(123,T_pump_in_1S + 273,Pressure_high_1S,Conc_pump_in_1S: T_pump_out_1S +
273,P_pump_out_1S,x_pump_out_1S,h_pump_out_1S,s_pump_out_1S,u_pump_out_1S,v_pump_out_1S,
q_pump_out_1S)
Q_pump_work1_1S = m_dot_high_concentration_1S * (h_pump_out_1S - h_pump_in_1S)
Q_pump_work2_1S = m_dot_high_concentration_1S * v_pump_in_1S * (Pressure_high_1S -
Pressure_low_1S) * 100
CALL
LiqLiqHeatExch(Pressure_high_1S,Pressure_low_1S,Conc_pump_in_1S,Conc_low_1S,eta_LLHX_1S,h_
pump_out_1S,h_gen_liq_out_1S
,m_dot_high_concentration_1S,m_dot_low_concentration_1S,T_pump_out_1S,T_gen_liq_out_1S:Q_LLH
X_1S,T_gen_in_1S,h_gen_in_1S,q_gen_in_1S,T_abs_liq_in_1S,h_abs_liq_in_1S,q_abs_liq_in_1S)
" Absorber Energy Requirements "
Q_absorber_DAY_1S = m_dot_low_conc_abs_DAY_1S*h_abs_liq_in_1S +
m_dot_Refrigerant_LPZ_Day_1S*h_abs_vap_in_1S -
m_dot_high_conc_abs_DAY_1S*h_abs_out_DAY_1S
Q_absorber_NIGHT_1S = m_dot_low_conc_abs_NIGHT_1S*h_abs_liq_in_1S +
m_dot_Refrigerant_LPZ_Night_1S*h_abs_vap_in_1S -
m_dot_high_conc_abs_NIGHT_1S*h_abs_out_NIGHT_1S
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" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high_1S,Conc_deph_out_1S,1: T_cond_in_1S +
273,P_cond_in_1S,x_cond_in_1S,h_cond_in_1S,s_cond_in_1S,u_cond_in_1S,v_cond_in_1S,q_cond_in_1
S)
" Condenser Energy Requirements "
Q_condenser_1S = m_dot_Refrigerant_HPZ_1S * (h_cond_in_1S - h_cond_out_1S)
" Generator Exit Conditions "
CALL
GeneratorLiquidHeating(Conc_pump_in_1S,Pressure_high_1S,Conc_low_1S,m_dot_high_concentration_
1S:T_gen_vap_out_1S,h_gen_vap_out_1S,x_gen_vap_out_1S)
CALL NH3H2O(238,Pressure_high_1S,Conc_pump_in_1S,0: T_gen_start_ACTUAL_1S +
273,P_gen_start_ACTUAL_1S,x_gen_start_ACTUAL_1S,h_gen_start_ACTUAL_1S,s_gen_start_ACTU
AL_1S,u_gen_start_ACTUAL_1S,v_gen_start_ACTUAL_1S,q_gen_start_ACTUAL_1S)
" Rectifier Analysis "
eta_Rectifier_1S = (T_gen_vap_out_1S - T_deph_in1_1S)/(T_gen_vap_out_1S - T_gen_in_1S)
" Principal Operating Line "
CALL NH3H2O(128,T_deph_in1_1S + 273,Pressure_high_1S,1: T_deph_in_1S +
273,P_deph_in_1S,x_deph_in_1S,h_deph_in_1S,s_deph_in_1S,u_deph_in_1S,v_deph_in_1S,q_deph_in_1
S)
Slope_1S = (h_deph_in_1S - h_gen_in_1S)/(x_deph_in_1S - Conc_pump_in_1S)
h_deph_in_1S = Slope_1S*x_deph_in_1S + constant_1S
" Generator Energy Requirements "
Pole_Generator_1S = Slope_1S*Conc_low_1S + constant_1S
Q_generator_1S = m_dot_low_concentration_1S * (h_gen_liq_out_1S - Pole_Generator_1S)
" Dephlegmator Energy Requirements "
Pole_Dephlegmator_1S = Slope_1S*Conc_deph_out_1S + constant_1S
Q_dephlegmator_1S = m_dot_Refrigerant_HPZ_1S * (Pole_Dephlegmator_1S - h_cond_in_1S)
" Coefficient of Performance "
COP_1S = Refrigeration_Effect_Day_1S/(Q_generator_1S + Q_pump_work2_1S)
" Energy Balance "
Energy_Balance_in_1S = (Refrigeration_Effect_Day_1S + Q_generator_1S +
Q_pump_work1_1S)*Hours_Day_1S
Energy_Balance_out_1S = Q_condenser_1S*Hours_Day_1S + Q_absorber_DAY_1S*Hours_Day_1S +
Q_absorber_NIGHT_1S*Hours_Night_1S + Q_dephlegmator_1S*Hours_Day_1S

"! ======================================= FIRST STAGE END
=========================================="
"! ======================================== OVERALL C.O.P.
=========================================="

COP_Overall = Refrigeration_Effect_Day_2S/(Q_generator_2S + Q_pump_work2_2S +Q_generator_1S +
Q_pump_work2_1S)
```

APPENDIX A6 (EES Module for Alternative Design D3)

```
T_gen_Day = 120
T_cond_Day = 45
T_evap_Day = -9
T_abs_Day = 45
Conc_deph_out_Day = 0.996
eta_LLHX_Day = 0.8
eta_VLHX_Day = 0.5
eta_Rectifier_Day = 0.75
Refrigeration_Effect_D&N = 5
Hours_Day = 24

" Required Refrigeration Effect"
Refrigeration_Effect_Day = Refrigeration_Effect_D&N*24/Hours_Day

" Parameters Selection "
CALL NH3H2O(138,T_evap_Day + 273,1,1: T_evap_out_Day +
273,P_evap_out_Day,x_evap_out_Day,h_evap_out_Day,s_evap_out_Day,u_evap_out_Day,v_evap_out_D
ay,q_evap_out_Day)
Pressure_low_Day = P_evap_out_Day
CALL NH3H2O(128,T_abs_Day +273,Pressure_low_Day,0: T_abs_out_Day
+273,P_abs_out_Day,x_abs_out_Day,h_abs_out_Day,s_abs_out_Day,u_abs_out_Day,v_abs_out_Day,q_a
bs_out_Day)
Conc_high_Day = x_abs_out_Day
CALL NH3H2O(138,T_cond_Day + 273,Conc_deph_out_Day,0: T_cond_out_Day +
273,P_cond_out_Day,x_cond_out_Day,h_cond_out_Day,s_cond_out_Day,u_cond_out_Day,v_cond_out_
Day,q_cond_out_Day)
Pressure_high_Day = P_cond_out_Day
CALL NH3H2O(128,T_gen_Day + 273,Pressure_high_Day,0: T_gen_liq_out_Day +
273,P_gen_liq_out_Day,x_gen_liq_out_Day,h_gen_liq_out_Day,s_gen_liq_out_Day,u_gen_liq_out_Day,v
_gen_liq_out_Day,q_gen_liq_out_Day)
Conc_low_Day = x_gen_liq_out_Day
CALL NH3H2O(238,Pressure_high_Day,Conc_high_Day,0: T_gen_start_Day +
273,P_gen_start_Day,x_gen_start_Day,h_gen_start_Day,s_gen_start_Day,u_gen_start_Day,v_gen_start_D
ay,q_gen_start_Day)
CALL NH3H2O(238,Pressure_low_Day,Conc_low_Day,0: T_abs_start_Day +
273,P_abs_start_Day,x_abs_start_Day,h_abs_start_Day,s_abs_start_Day,u_abs_start_Day,v_abs_start_Da
y,q_abs_start_Day)

" Vapor Liquid Heat Exchanger "
eta_VLHX_Day = (T_vap_out_VLHX_Day - T_evap_out_Day)/(T_cond_out_Day - T_evap_out_Day)
CALL NH3H2O(123,T_vap_out_VLHX_Day + 273,Pressure_low_Day,1: T_abs_vap_in_Day +
273,P_abs_vap_in_Day,x_abs_vap_in_Day,h_abs_vap_in_Day,s_abs_vap_in_Day,u_abs_vap_in_Day,v_a
bs_vap_in_Day,q_abs_vap_in_Day)
h_liq_out_VLHX_Day = h_cond_out_Day - (h_abs_vap_in_Day - h_evap_out_Day)
CALL NH3H2O(234,Pressure_high_Day,Conc_deph_out_Day,h_liq_out_VLHX_Day:
T_before_expansion_Day +
273,P_before_expansion_Day,x_before_expansion_Day,h_before_expansion_Day,s_before_expansion_Da
y,u_before_expansion_Day,v_before_expansion_Day,q_before_expansion_Day)

" Ammonia Expansion Process "
CALL NH3H2O(234,Pressure_low_Day,Conc_deph_out_Day,h_before_expansion_Day:
T_after_expansion_Day +
```

```
273,P_after_expansion_Day,x_after_expansion_Day,h_after_expansion_Day,s_after_expansion_Day,u_after_expansion_Day,v_after_expansion_Day,q_after_expansion_Day)

" Refrigerant Flow Rate"
Refrigeration_Effect_Day = m_dot_Refrigerant_Day * (h_evap_out_Day - h_after_expansion_Day)

" Energy Exchange at VLHX "
Q_VLHX1_Day = m_dot_Refrigerant_Day * (h_cond_out_Day - h_liq_out_VLHX_Day)
Q_VLHX2_Day = m_dot_Refrigerant_Day * (h_abs_vap_in_Day - h_evap_out_Day)

" High and Low Concentration Mass Flow Rates "
m_dot_low_concentration_Day + m_dot_Refrigerant_Day = m_dot_high_concentration_Day
m_dot_low_concentration_Day*Conc_low_Day + m_dot_Refrigerant_Day*Conc_deph_out_Day =
m_dot_high_concentration_Day*Conc_high_Day

" Pump Work "
CALL NH3H2O(123,T_abs_out_Day + 273,Pressure_high_Day,Conc_high_Day: T_pump_out_Day +
273,P_pump_out_Day,x_pump_out_Day,h_pump_out_Day,s_pump_out_Day,u_pump_out_Day,v_pump_
out_Day,q_pump_out_Day)
Q_pump_work1_Day = m_dot_high_concentration_Day * (h_pump_out_Day - h_abs_out_Day)
Q_pump_work2_Day = m_dot_high_concentration_Day * v_abs_out_Day * (Pressure_high_Day -
Pressure_low_Day) * 100

CALL
LiqLiqHeatExch(Pressure_high_Day,Pressure_low_Day,Conc_high_Day,Conc_low_Day,eta_LLHX_Day,
h_pump_out_Day,h_gen_liq_out_Day
,m_dot_high_concentration_Day,m_dot_low_concentration_Day,T_pump_out_Day,T_gen_liq_out_Day:Q
_LLHX_Day,T_gen_in_Day,h_gen_in_Day,q_gen_in_Day,T_abs_liq_in_Day,h_abs_liq_in_Day,q_abs_li
q_in_Day)


" Absorber Energy Requirements "
Q_absorber_Day = m_dot_low_concentration_Day*h_abs_liq_in_Day +
m_dot_Refrigerant_Day*h_abs_vap_in_Day - m_dot_high_concentration_Day*h_abs_out_Day

" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high_Day,Conc_deph_out_Day,1: T_cond_in_Day +
273,P_cond_in_Day,x_cond_in_Day,h_cond_in_Day,s_cond_in_Day,u_cond_in_Day,v_cond_in_Day,q_c
ond_in_Day)

" Condenser Energy Requirements "
Q_condenser_Day = m_dot_Refrigerant_Day * (h_cond_in_Day - h_cond_out_Day)

" Generator Exit Conditions "
CALL
GeneratorLiquidHeating(Conc_high_Day,Pressure_high_Day,Conc_low_Day,m_dot_high_concentration_
Day:T_gen_vap_out_Day,h_gen_vap_out_Day,x_gen_vap_out_Day)

" Rectifier Analysis "
eta_Rectifier_Day = (T_gen_vap_out_Day - T_deph_in1_Day)/(T_gen_vap_out_Day - T_gen_in_Day)

" Principal Operating Line "
CALL NH3H2O(128,T_deph_in1_Day + 273,Pressure_high_Day,1: T_deph_in_Day +
273,P_deph_in_Day,x_deph_in_Day,h_deph_in_Day,s_deph_in_Day,u_deph_in_Day,v_deph_in_Day,q_d
eph_in_Day)
Slope_Day = (h_deph_in_Day - h_gen_in_Day)/(x_deph_in_Day - Conc_high_Day)
```

```
h_deph_in_Day = Slope_Day*x_deph_in_Day + constant_Day

" Generator Energy Requirements "
Pole_Generator_Day = Slope_Day*Conc_low_Day + constant_Day
Q_generator_Day = m_dot_low_concentration_Day * (h_gen_liq_out_Day - Pole_Generator_Day)

" Dephlegmator Energy Requirements "
Pole_Dephlegmator_Day = Slope_Day*Conc_deph_out_Day + constant_Day
Q_dephlegmator_Day = m_dot_Refrigerant_Day * (Pole_Dephlegmator_Day - h_cond_in_Day)

" Coefficient of Performance "
COP_Day = Refrigeration_Effect_Day/(Q_generator_Day + Q_pump_work2_Day)

" Energy Balance "
Energy_Balance_in_Day = Refrigeration_Effect_Day + Q_generator_Day + Q_pump_work1_Day
Energy_Balance_out_Day = Q_condenser_Day + Q_absorber_Day + Q_dephlegmator_Day


" Units deg C, bar, kJ, kg, m, sec, kW "
" sc  = solar collector  "
" ********************************  Input Data
************************************************** "
T_gen_Night = 112
T_cond_Night = 37
T_evap_Night = -9
T_abs_Night = 37
Conc_deph_out_Night = 0.996
eta_LLHX_Night = 0.8
eta_VLHX_Night = 0.5
eta_Rectifier_Night = 0.75
{Refrigeration_Effect_D&N_Night = 5}
Hours_Night = 24

" ******************************** SOLUTION
**************************************************"

" Required Refrigeration Effect"
Refrigeration_Effect_Night = Refrigeration_Effect_D&N*24/Hours_Night

" Parameters Selection "
CALL NH3H2O(138,T_evap_Night + 273,1,1: T_evap_out_Night +
273,P_evap_out_Night,x_evap_out_Night,h_evap_out_Night,s_evap_out_Night,u_evap_out_Night,v_evap
_out_Night,q_evap_out_Night)
Pressure_low_Night = P_evap_out_Night
CALL NH3H2O(128,T_abs_Night +273,Pressure_low_Night,0: T_abs_out_Night
+273,P_abs_out_Night,x_abs_out_Night,h_abs_out_Night,s_abs_out_Night,u_abs_out_Night,v_abs_out_
Night,q_abs_out_Night)
Conc_high_Night = x_abs_out_Night
CALL NH3H2O(138,T_cond_Night + 273,Conc_deph_out_Night,0: T_cond_out_Night +
273,P_cond_out_Night,x_cond_out_Night,h_cond_out_Night,s_cond_out_Night,u_cond_out_Night,v_con
d_out_Night,q_cond_out_Night)
Pressure_high_Night = P_cond_out_Night
CALL NH3H2O(128,T_gen_Night + 273,Pressure_high_Night,0: T_gen_liq_out_Night +
273,P_gen_liq_out_Night,x_gen_liq_out_Night,h_gen_liq_out_Night,s_gen_liq_out_Night,u_gen_liq_out_
Night,v_gen_liq_out_Night,q_gen_liq_out_Night)
Conc_low_Night = x_gen_liq_out_Night
```

```
CALL NH3H2O(238,Pressure_high_Night,Conc_high_Night,0: T_gen_start_Night +
273,P_gen_start_Night,x_gen_start_Night,h_gen_start_Night,s_gen_start_Night,u_gen_start_Night,v_gen_
start_Night,q_gen_start_Night)
CALL NH3H2O(238,Pressure_low_Night,Conc_low_Night,0: T_abs_start_Night +
273,P_abs_start_Night,x_abs_start_Night,h_abs_start_Night,s_abs_start_Night,u_abs_start_Night,v_abs_st
art_Night,q_abs_start_Night)

" Vapor Liquid Heat Exchanger "
eta_VLHX_Night = (T_vap_out_VLHX_Night - T_evap_out_Night)/(T_cond_out_Night -
T_evap_out_Night)
CALL NH3H2O(123,T_vap_out_VLHX_Night + 273,Pressure_low_Night,1: T_abs_vap_in_Night +
273,P_abs_vap_in_Night,x_abs_vap_in_Night,h_abs_vap_in_Night,s_abs_vap_in_Night,u_abs_vap_in_Ni
ght,v_abs_vap_in_Night,q_abs_vap_in_Night)
h_liq_out_VLHX_Night = h_cond_out_Night - (h_abs_vap_in_Night - h_evap_out_Night)
CALL NH3H2O(234,Pressure_high_Night,Conc_deph_out_Night,h_liq_out_VLHX_Night:
T_before_expansion_Night +
273,P_before_expansion_Night,x_before_expansion_Night,h_before_expansion_Night,s_before_expansio
n_Night,u_before_expansion_Night,v_before_expansion_Night,q_before_expansion_Night)

" Ammonia Expansion Process "
CALL NH3H2O(234,Pressure_low_Night,Conc_deph_out_Night,h_before_expansion_Night:
T_after_expansion_Night +
273,P_after_expansion_Night,x_after_expansion_Night,h_after_expansion_Night,s_after_expansion_Night
,u_after_expansion_Night,v_after_expansion_Night,q_after_expansion_Night)

" Refrigerant Flow Rate"
Refrigeration_Effect_Night = m_dot_Refrigerant_Night * (h_evap_out_Night - h_after_expansion_Night)

" Energy Exchange at VLHX "
Q_VLHX1_Night = m_dot_Refrigerant_Night * (h_cond_out_Night - h_liq_out_VLHX_Night)
Q_VLHX2_Night = m_dot_Refrigerant_Night * (h_abs_vap_in_Night - h_evap_out_Night)

" High and Low Concentration Mass Flow Rates "
m_dot_low_concentration_Night + m_dot_Refrigerant_Night = m_dot_high_concentration_Night
m_dot_low_concentration_Night*Conc_low_Night + m_dot_Refrigerant_Night*Conc_deph_out_Night =
m_dot_high_concentration_Night*Conc_high_Night

" Pump Work "
CALL NH3H2O(123,T_abs_out_Night + 273,Pressure_high_Night,Conc_high_Night: T_pump_out_Night
+
273,P_pump_out_Night,x_pump_out_Night,h_pump_out_Night,s_pump_out_Night,u_pump_out_Night,v_
pump_out_Night,q_pump_out_Night)
Q_pump_work1_Night = m_dot_high_concentration_Night * (h_pump_out_Night - h_abs_out_Night)
Q_pump_work2_Night = m_dot_high_concentration_Night * v_abs_out_Night * (Pressure_high_Night -
Pressure_low_Night) * 100

CALL
LiqLiqHeatExch(Pressure_high_Night,Pressure_low_Night,Conc_high_Night,Conc_low_Night,eta_LLHX
_Night,h_pump_out_Night,h_gen_liq_out_Night
,m_dot_high_concentration_Night,m_dot_low_concentration_Night,T_pump_out_Night,T_gen_liq_out_Ni
ght:Q_LLHX_Night,T_gen_in_Night,h_gen_in_Night,q_gen_in_Night,T_abs_liq_in_Night,h_abs_liq_in_
Night,q_abs_liq_in_Night)


" Absorber Energy Requirements "
```

```
Q_absorber_Night = m_dot_low_concentration_Night*h_abs_liq_in_Night +
m_dot_Refrigerant_Night*h_abs_vap_in_Night - m_dot_high_concentration_Night*h_abs_out_Night

" Dephlegmator Exit Temperature "
CALL NH3H2O(238,Pressure_high_Night,Conc_deph_out_Night,1: T_cond_in_Night +
273,P_cond_in_Night,x_cond_in_Night,h_cond_in_Night,s_cond_in_Night,u_cond_in_Night,v_cond_in_
Night,q_cond_in_Night)

" Condenser Energy Requirements "
Q_condenser_Night = m_dot_Refrigerant_Night * (h_cond_in_Night - h_cond_out_Night)

" Generator Exit Conditions "
CALL
GeneratorLiquidHeating(Conc_high_Night,Pressure_high_Night,Conc_low_Night,m_dot_high_concentrati
on_Night:T_gen_vap_out_Night,h_gen_vap_out_Night,x_gen_vap_out_Night)

" Rectifier Analysis "
eta_Rectifier_Night = (T_gen_vap_out_Night - T_deph_in1_Night)/(T_gen_vap_out_Night -
T_gen_in_Night)

" Principal Operating Line "
CALL NH3H2O(128,T_deph_in1_Night + 273,Pressure_high_Night,1: T_deph_in_Night +
273,P_deph_in_Night,x_deph_in_Night,h_deph_in_Night,s_deph_in_Night,u_deph_in_Night,v_deph_in_
Night,q_deph_in_Night)
Slope_Night = (h_deph_in_Night - h_gen_in_Night)/(x_deph_in_Night - Conc_high_Night)
h_deph_in_Night = Slope_Night*x_deph_in_Night + constant_Night

" Generator Energy Requirements "
Pole_Generator_Night = Slope_Night*Conc_low_Night + constant_Night
Q_generator_Night = m_dot_low_concentration_Night * (h_gen_liq_out_Night - Pole_Generator_Night)

" Dephlegmator Energy Requirements "
Pole_Dephlegmator_Night = Slope_Night*Conc_deph_out_Night + constant_Night
Q_dephlegmator_Night = m_dot_Refrigerant_Night * (Pole_Dephlegmator_Night - h_cond_in_Night)

" Coefficient of Performance "
COP_Night = Refrigeration_Effect_Night/(Q_generator_Night + Q_pump_work2_Night)

" Energy Balance "
Energy_Balance_in_Night = Refrigeration_Effect_Night + Q_generator_Night + Q_pump_work1_Night
Energy_Balance_out_Night = Q_condenser_Night + Q_absorber_Night + Q_dephlegmator_Night


Solar_Available_Time = 10
Q_Heat_Storage = Q_generator_Night * (24 - Solar_Available_Time) / Solar_Available_Time

Pressure = 12
Storage_Heat =(Q_Heat_Storage * 3600 * Solar_Available_Time) / (Enthalpy(Water,T=T_gen_Night + 40
,P=Pressure) - Enthalpy(Water,T=T_gen_Night ,P=Pressure))
A = Quality(Water,T=T_gen_Night + 40 ,P=Pressure)
```

NOMENCLATURE

$Conc_{deph,out}$	Ammonia Mass Concentration at the exit of Dephlegmator
$Conc_{high,DAY}$	High Concentration of Ammonia in the system (Day time)
$Conc_{high,NIGHT}$	High Concentration of Ammonia in the system (Night time)
$Conc_{low}$	Low Concentration of Ammonia in the system
$Conc_{pump,in}$	Ammonia Concentration at the pump inlet
COP	Coefficient of Performance
$E_{absorption}$	Energy rejected during absorption process (kJ)
$E_{condensation}$	Energy rejected during condensation process (kJ)
$E_{Depressurization}$	Energy rejected during depressurization process (kJ)
$E_{generation}$	Energy supplied during generation process (kJ)
$E_{pressurization}$	Energy supplied during pressurization process (kJ)
E_{VLHX}	Energy transfer by Vapor Liquid Heat Exchanger (kJ)
$h_{abs,vap,in}$	Enthalpy of ammonia vapor at the inlet of absorber (kJ/kg)
$h_{after,expansion}$	Enthalpy across expansion valve (kJ/kg)
$h_{cond,out}$	Enthalpy at the exit of condenser (kJ/kg)
$h_{evap,out}$	Enthalpy at the exit of evaporator (kJ/kg)

h_{l}	Enthalpy of Saturated Liquid Aqua-Ammonia (kJ/kg)
$h_{liq,out,VLHX}$	Enthalpy of Sub cooled Ammonia at the exit of VLHX (kJ/kg)
$h_{pump,in}$	Enthalpy of Aqua-Ammonia at the pump inlet (kJ/kg)
h_{v}	Enthalpy of Saturated Vapor Aqua-Ammonia (kJ/kg)
$Hours_{Day}$	Available Sunlight hours
$Hours_{Night}$	Unavailability of Sunlight hours
$ICE_{Storage}$	Mass of Ice Storage system (kg)
$m_{high,conc,strongsol}$	Mass flow rate of high conc leaving strong sol tank (kg/s)
$m_{low,conc,weaksol}$	Mass flow rate of low conc entering weak sol tank (kg/s)
$m_{Refrigerant,HPZ}$	Mass flow rate of the refrigerant in the high pressure zone (kg/s)
$m_{Refrigerant,LPZ}$	Mass flow rate of the refrigerant in the low pressure zone (kg/s)
$Mass_{Refrigerant}$	Total mass of the refrigerant required for refrigeration (kg)
n_{LLHX}	Effectiveness of Liquid-Liquid Heat Exchanger
n_{VLHX}	Effectiveness of Vapor-Liquid Heat Exchanger
$P_{cond,out}$	Condenser Pressure (bar)
$P_{evap,out}$	Evaporator Pressure (bar)
$Pressure_{high}$	High pressure in the system (bar)

$Pressure_{low}$	Low pressure in the system (bar)
$Q_{absorber,DAY}$	Heat Duty of the Absorber (Day Time) (kW)
$Q_{absorber,NIGHT}$	Heat Duty of the Absorber (Night Time) (kW)
$Q_{condenser}$	Heat Duty of the Condenser (kW)
$Q_{Dephlegmator}$	Heat Duty of the Dephlegmator (kW)
$Q_{evaporator}$	Heat Duty of the Evaporator (kW)
$Q_{Generator}$	Heat Duty of the Generator (kW)
Q_{LLHX}	Heat Duty of the LLHX (kW)
$Q_{pump,work}$	Heat Duty of the Pump (kW)
Q_{VLHX}	Heat Duty of the VLHX (kW)
$Refrigeration_{Effect,D\&N}$	Complete 24 hours Refrigeration Effect required (kW)
$Refrigeration_{Effect,Day}$	Day Time Refrigeration Effect Required (kW)
$Storage_{Heat}$	Mass of Heat Storage system (kg)
$Storage_{Liquid,Ammonia}$	Mass of Liquid Ammonia Storage system (kg)
$Storage_{Strong,Solution}$	Mass of Strong Solution Storage system (kg)
$Storage_{Weak,Solution}$	Mass of Weak Solution Storage system (kg)
$T_{cond,out}$	Temperature at the exit of condenser (^{o}C)

$T_{deph,in}$	Temperature at the inlet of dephlegmator (ºC)
$T_{evap,out}$	Temperature at the exit of evaporator (ºC)
$T_{gen,in}$	Temperature at the inlet of generator (ºC)
$T_{gen,vap,out}$	Temperature of ammonia vapor at the exit of generator (ºC)
T_{l}	Temperature of Saturated Aqua-Ammonia liquid (ºC)
T_{v}	Temperature of Saturated Aqua-Ammonia vapor (ºC)
$T_{vap,out,VLHX}$	Temperature of vapor at the exit of VLHX (ºC)
$x_{abs,out,DAY}$	Ammonia Mass Concentration at the exit of absorber (Day time)
$x_{abs,out,NIGHT}$	Ammonia Mass Concentration at the exit of absorber (Night time)
$x_{gen,liq,out}$	Ammonia Mass Concentration of the liquid at the exit of generator

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